

**DIESEL ENGINE OPERATION
ON ALCOHOL FUELS USING
A COMPUTER CONTROLLED FUMIGATION PROCESS**

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NOMENCLATURE

h	Air flow rate meter reading (cm)
K	Temperature compensation factor
M	Mass flow rate (kg/hr)
N	Engine speed (rpm)
S	Spring balance force (N)
T	Temperature (°C)
V	Volume flow rate (m ³ /hr)
W	Balance weight (kg)
η	efficiency
λ	Equivalence ratio

SUBSCRIPTS

a	air
a.f.	<u>m</u> aximum <u>f</u> umigation
atmos	atmospheric
f	fuel
fumi	fumigation
i.f.	<u>m</u> inimum <u>f</u> umigation
in	inlet air
ind	indicated
inj	injection
m	(inlet) manifold
p	pressure
th	thermal
vol	volume

ABBREVIATIONS

AC	Alternative current
----	---------------------

A/D	Analog/Digital
BDC	Bottom dead centre
BMEP	Brake mean effective pressure
Bp	Brake power
BTDC	Before top dead centre
CI	Compression ignition
CJC	Cold junction temperature
CNG	Compressed natural gas
CO	Carbon monoxide
DC	Direct current
DI	Direct Injection
E	Earth
FC	Fuel consumption
HC	Hydrocarbon
I/O	Input/output
IDI	Indirect injection
LCV	Lower calorific value
LFTB	Liquid Fuel Trust Board
LPG	Liquified petroleum gas
MTG	Methanol to gasoline
NO _x	Total oxides of nitrogen
P	Phase
ppm	parts per million
rpm	revolutions per minute
SG	Specific gravity
SCR	Silicon controlled rectifier
TDC	Top dead centre
TTL	Transistor-Transistor Logic

ABSTRACT

Alcohol fuels have been shown to be very attractive alternative fuels for automotive use. They are considered to be cleaner fuels for protection of the environment and they are good alternatives to petrol because of higher octane numbers. Use of alcohol fuels in compression ignition engines has also attracted worldwide interest due mainly to its ability to burn without producing particulate emissions.

This study shows that an inexpensive conversion system developed on a small single cylinder research engine enables a diesel engine to operate on near 100 percent methanol or ethanol. A computer control system has been built to exploit the possibility of on board vehicle application in the future.

A fumigation method was adopted for this work, which involved adding a carburettor to the inlet manifold to fumigate the inlet air with alcohol fuel. This mixture formed an ignition centre for the main body of alcohol fuel injected into the cylinder in a conventional manner. A computer acquisition system enabled the fumigated mixture to be controlled at a suitable percentage of alcohol and inlet temperature, which helped the evaporation of induced alcohol, to produce the optimal engine performance under the specific speed and load conditions.

Test results showed the satisfactory combustion over

the range of engine operating conditions. This method has the potential of allowing diesel engines to operate on virtually 100% alcohol without considerable change to the engine and hopefully to the road use after further investigations.

CHAPTER I

INTRODUCTION

Since the "oil crisis" of the early 1970's, alternative engine fuels have been investigated and developed in many countries for both commercial and political reasons. New Zealand suffered badly during the oil crisis because more than 90 percent of its transport fuel requirement was imported at that time. The Liquid Fuel Trust Board, which was funded by the government, undertook management of research into alternative fuels and helped develop a strategy for the fuel industry. Our interest and the scope of this research project concerning on alternative fuels for diesel engines comes from a review of the situation of fuel industry in New Zealand, technical feasibility study and potential practical applications.

I. FUEL INDUSTRY IN NEW ZEALAND

Prior to the oil embargo in the Middle East, most of our transportation fuels were dependant on the importation of oil into New Zealand. The prospect of using natural gas from the large Maui gas field (discovered in 1969) to lessen the economic impact of the energy crisis led to the building of the world first synthetic fuel plant using the Mobil process, which processes natural gas from the Maui and Kapuni fields into synthetic gasoline. A two stage process is utilised where natural gas is converted to

methanol, and then converted into synthetic petrol using the Mobil patented ZSM-5 catalyst. At the design production of 570,000 tonnes per annum, the complex produces the equivalent of approximately 1/3 of New Zealand's petrol requirement.

It is quite expensive to convert methanol to synthetic petrol with the above process, hence alternative methods should be investigated. Direct use of methanol as a transportation fuel, replacing petrol, has been undertaken for a long time with the aim being to clean up vehicle exhaust emissions. In the United States, there is a strong move to utilise pure methanol as a fuel and this has been shown to reduce vehicle emissions of hydrocarbons (unburnt fuel) and the greenhouse gases. Success in this area would also lessen U.S. dependence on foreign energy sources. Meanwhile in Brazil, ethanol has become a very popular substitution for petrol to help solve the foreign exchange problem and make full use of domestic agricultural resources.

Table 1.1 shows the level of fuel self sufficiency in petrol and diesel engines in New Zealand. Increasing self sufficiency in petrol from sources other than crude oil will cause an even greater imbalance in production of diesel oil. The high cost of diesel has an impact on the cost of almost all commodities to which transportation plays a vital role, so any fuel cost savings that can be made will be of benefit to all areas of the economy. This highlights the importance of investigating alternative fuels for diesel engines.

	Petrol fuel (Otto cycle engine)	Diesel fuel (CI cycle engine)
Demand	1,800,000 tonnes	1,100,000 tonnes
N.Z. oil and condensate	675,000	380,000
CNG and LPG	155,000	-
synthetic petrol	570,000	-
Self sufficiency	77%	35%

Table 1.1 New Zealand's level of fuel self sufficiency

II. ALTERNATIVE FUELS FOR DIESEL ENGINE

There are several alternative fuels for diesel engines, such as compressed natural gas (CNG), liquified petroleum gas (LPG), the alcohols, diesel blends and extenders. These possible diesel alternatives could come from a number of natural resources. Figure 1.1 (1) shows the different process which are feasible to convert the various raw energy resources into transport fuels in New Zealand.

The following reviews some of the more promising alternative replacement fuels for diesel:

(1) Compressed Natural Gas

CNG cannot be used directly in a diesel engine, some necessary changes must be made to the engine before its utilisation. There are two general accepted ways:

(a) Spark Ignition Operation (2)

This method requires converting the diesel engine to spark ignition operation. This includes removing the injectors and diesel pump, and modifying the combustion

chamber to get a lower compression ratio suitable for ignition without pre-ignition or auto-ignition.

Engine efficiency has been shown to be almost the same as that of diesel operation, although an increase may be noticed at low engine speeds and a decrease at higher engine speed (3). It is costly to convert using this method and very expensive to convert back to diesel once a change has been made. A major disadvantage of vehicle operation with this type of conversion is the weight of cylinders which are used to carry CNG. Only partial compensation could be made by removing the diesel tank.

(b) Dual Fuel Operation (3,4,5,6)

In this method, a small percentage of diesel fuel is used to establish a pilot ignition for CNG. It has the desirable characteristic that the vehicle can easily revert to diesel operation, thereby enabling its use in areas remote from CNG supply. On the other hand, the proportion of CNG and diesel which can be accommodated by the engine varies significantly with engine load. The disadvantage of dual fuel CNG/diesel operation is that it does not allow achievement of full cost advantages of CNG use since some diesel is always used.

(2) Liquified Petroleum Gas

Similar to the CNG in operation, LPG can be used in diesel engine in two ways:

Converting diesel engine to spark ignition.

Dual fuel operation.

Because of the similarity in combustion characteristics between LPG and premium petrol, the most

accepted method for its use is to convert the diesel engine to spark ignition and Otto cycle operation. The technology for such conversion has been widely explored overseas (7).

(3) Alcohol Fuels (8)

The alcohol fuels are seen as being able to offer a solution to a variety of problems faced by various countries. Such problems range from balance of payment difficulties when importing overseas oil, to pollution reduction from engines in those countries enforcing stricter emission control laws, such as United States.

Alcohols as alternative fuel for diesel engines has brought about world-wide interest. Conversion procedures from diesel to alcohol fuels may be conveniently subdivided into two groups. One involves the fuel management procedures in which changes are required to the fuel itself, another involves engine management procedures which require changes to the engine. A more detailed description of these procedures will be displayed later in the thesis.

(4) Other Fuels

Biomass derived fuels, comprising natural fats and oil (such as sunflower oil), may be used in unprocessed form in diesel engines. However, biomass feedstock are inherently disadvantaged when compared to fossil feedstock because of the variety of feedstock, handling costs, multiple markets and multiple ownership (9). Technical problems such as high viscosity remains unsolved. Tallow ester as an alternative diesel fuel in New Zealand may be

feasible because the above disadvantages of biomass do not in general apply to the tallow resource.

Biogas has been recognized as a promising alternative in view of its lower cost of production since it is normally produced from organic wastes in the presence of bacteria, thus it is freely available in nature. It has been reported to achieve a lower efficiency at lower loads and higher efficiency at higher engine power when compared to the diesel operation. Biogas operation reduces the exhaust gas temperature and also nitric oxide emissions, but excess amount of carbon dioxide has been noted. Storage is another problem (10).

Other alternative fuels for diesel substitution under investigation are: ammonia, low octane gasoline, kerosine, hydrogen, tar creosote and coal dust/oil slurries (11).

III. SCOPE OF THIS RESEARCH

From figure 1.1, alcohol fuels, both methanol and ethanol, can be made available in New Zealand through various resources. New Zealand has the first Mobil Methanol-to-Gasoline (MTG) process in the world to produce synthetic petrol. MTG production involves the intermediate formation of methanol from synthesis gas, then methanol vapour is contacted with a patented catalyst which eliminates water and produces a range of liquid hydrocarbons.

Besides natural gas, coal, peat and wood can also be

used as a resource to produce methanol, using the different technologies displayed in figure 1.1.

Ethanol can be produced from abundant resources available in New Zealand, such as wood and agricultural biomass, e.g. sugarbeet, fodder beet, maize and protein extracted forage (lucerne).

Alcohol fuels have such a great prospect use as automobile fuels that the scope of this research involves the application of alcohol fuels for diesel engine operation.

IV. LAY-OUT OF THESIS

This thesis highlights various methods which can use the alcohols as alternative fuels for diesel engines. Tests on a single cylinder diesel engine operating on either methanol or ethanol, by means of a simple fumigation system, are presented. The design, building and testing of a computer data acquisition and control system to monitor and control the fumigation system is also given. Finally the results are discussed and conclusions and recommendations presented.

All the results, together with the discussions, conclusions and recommendations, are presented in the thesis as follows:

Chapter 1 introduces the situation regarding industrial fuel usage in New Zealand, the available resources to produce alternative fuels for transport and conversion techniques for these prospective alternative fuels. Alcohol fuels, methanol and ethanol, can be

produced from abundant resources available in New Zealand, Therefore they are the focus of this study.

Chapter 2 summarises research concerning alcohols as automotive fuels. A detailed literature review of alcohol fuels used in diesel engines is presented. The fumigation method seems to be an inexpensive way to convert diesel engine to operate on alcohol. Early research results also encouraged further development of this method. Therefore this study has concentrated on this technique.

Chapter 3, a brief introduction to compression ignition combustion and the effect of fumigation on the combustion process is presented. Next both diesel and alcohol fuel characteristics are compared. Their combustion characteristics are also described in order to prepare for a better theoretical understanding of the tests to follow.

Chapter 4 describes the modifications made to a single cylinder research engine to allow alcohol fuel to fumigate with the inlet air via a small jetted carburettor. An introduction to the test equipment is also presented.

Chapter 5 presents the baseline test results and discussion. The calculation equations for the test engine are also listed. Graphical displays of engine performance on both diesel and alcohol fuels are presented. Sections 5.3 to 5.6 discuss the possible fumigation range and the relative air/fuel ratios which are lean enough not to cause auto-ignition during the compression stroke prior to fuel injection. Details of auxiliary heating is also presented. Analysis of combustion characteristics and ignition delay is presented in sections 5.7 and 5.8. Section 5.9

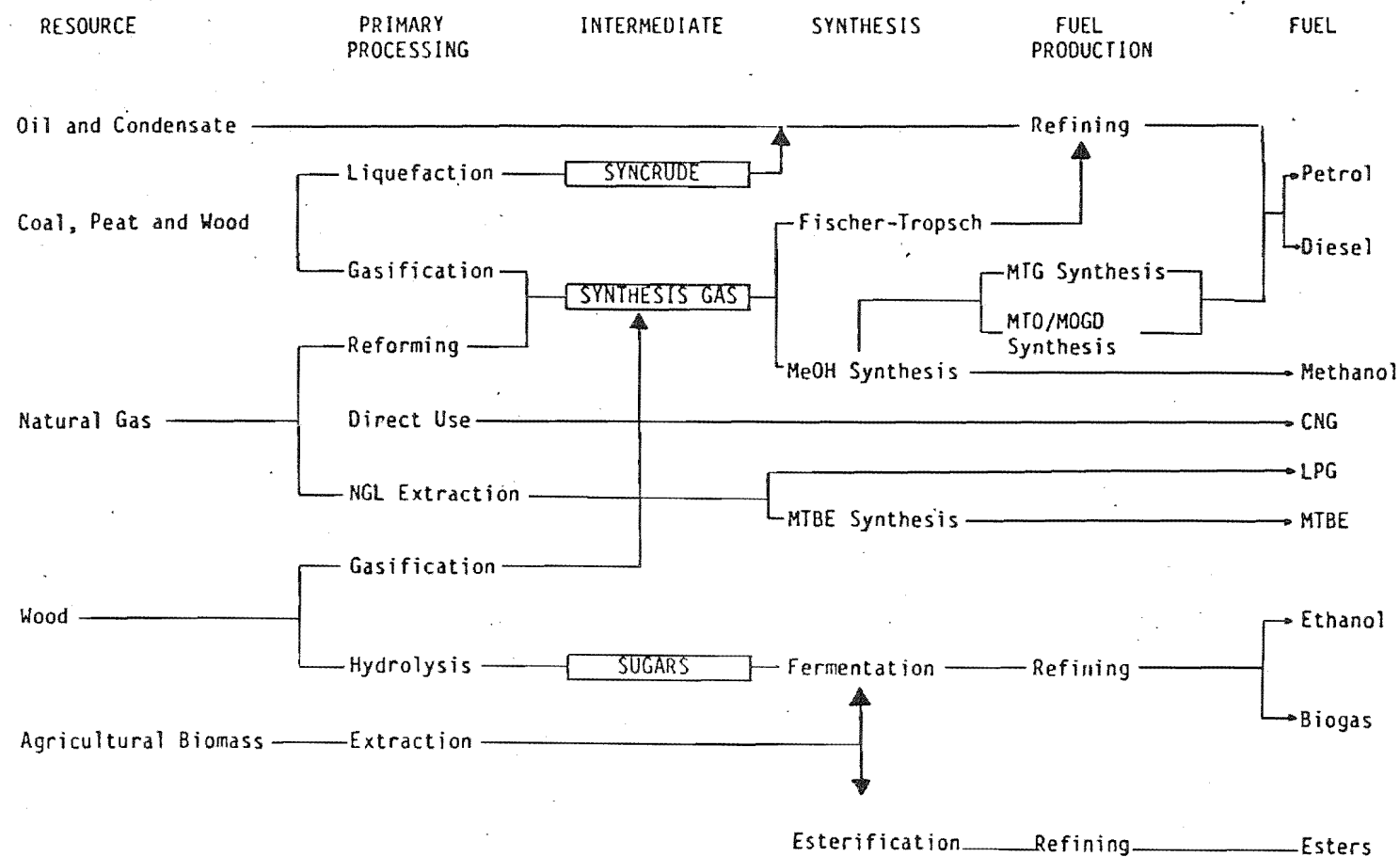
describes the effect on performance of different engine set-ups. Finally a preliminary discussion is given which also provides details for the development of a computer controlled system.

Chapter 6 is based on the baseline test data. It presents all the hardware developed for the control system as well as software concepts. Section 6.1 summarises all the test signals. Sections 6.2 and 6.3 introduce the establishment of a data acquisition and control system. The final section outlines the control programme "MODE-T".

Chapter 7 presents results from the operation of the control system and related discussions. Presented first is a summary of test results, this is compared with baseline tests and finally conclusions for this control system are presented.

Chapter 8, all the test results are summarised and the final conclusion from this study is presented. The final section gives recommendations for future work.

FIGURE 1.1 FUEL PRODUCTION ROUTES



CHAPTER II

LITERATURE REVIEW OF ALCOHOL FUELS

I. HISTORICAL REVIEW OF ALCOHOL FUELS

Alcohol used as a fuel goes back over a century. The most popular two alcohol fuels are methanol and ethanol.

Methanol is a clean synthetic fuel and chemical feedstock, which can be made from a wide variety of renewable as well as conventional materials and energy sources, it is applied to an equally wide variety of use (12). The known uses of methanol as a fuel in early time was connected with the production of charcoal by condensing the vapours driven off. By the middle of the 19th century, methanol was well established in France as a clean fuel for cooking, heating and lighting until replaced by kerosene around 1888. Methanol was commonly called wood alcohol because it was once exclusively produced by the destructive distillation of wood. Today it is widely used as racing fuel, where economy is not considered so important.

Ethanol can be produced by biomass conversion pathways. The sugar and starches of agricultural crops can be easily converted to ethanol by anaerobic fermentation. Cellulose, together with hemi-cellulose and lignin, are the major constituents of wood. The cellulose and hemi-cellulose are polysaccharides which are complex polymers of

sugars, and can be used as feedstock for ethanol production, provided that they can be hydrolysed to simple sugars.

Here in New Zealand, a significant fraction of transport fuel supply is now produced in the form of methanol. Most of this (4,400 tonnes/day) is converted to synthetic petrol using the expensive Mobil (MTG) process. Large natural resources could be developed to supply a methanol-based transport fuels economy. The Liquid Fuels Trust Board, which was funded by the New Zealand Government, took an active role in the field of methanol use as a substitute for petrol and to lessen extent diesel in this country. The status of technology regarding the use of alcohol fuels in engines as researched by the LFTB is outlined briefly as following (1):

(a) Alcohol Blends with Petrol

The use of both methanol and ethanol as petrol extenders has been established for a number of years in several countries. Ethanol has been used as a petrol blendstock in Brazil and USA, and also introduced in a number of developing countries such as Zimbabwe and Malawi. While methanol is in common use as a petrol blendstock in Germany and USA.

(b) Alcohol Emulsions with Diesel

The blending of methanol at the low levels was considered uneconomic because of the technical disadvantages and the high cost of alcohol co-solvents (1). The work was ceased on low methanol blends in diesel cycle engine, in which there was 15 percent or 20 percent

emulsions, because of the vapour locks caused by the fuel separation.

(c) High Percentage Alcohol Fuelling as Petrol Substitute

Conversion systems have been developed to run petrol engine fuelled by 100% methanol. Cold starting and cold drivability were identified as areas requiring improvement.

(d) High Percentage Alcohol Fuelling as Diesel Substitute

Solutions to the high percentage methanol blends that have been most fully developed include:

Spark ignited diesel cycle

Dual-fuel injection

Otto-cycle conversion

Chemical ignition improvement

Of all the alternative fuels investigated by LFTB, methanol appears to have the best potential for use as either a petrol or diesel replacement (13). The importance of methanol as a diesel substitute was that there were comparatively few really viable options for diesel substitution, even though the need for this was of considerable national importance in order to redress the emerging petro/diesel substitution imbalance. Also most essential commercial transport operations use diesel rather than petrol-fuelled vehicles. So the alcohols, especially methanol, is the major fuel considered in this study.

II. ALCOHOLS AS ALTERNATIVE FUELS FOR CI ENGINE

The alcohols are favourable alternative fuels for

petrol engines. On the other hand, they are difficult to be used in CI engine because of their low cetane rating which signifies a long ignition delay period. To try and overcome this problem, a great deal of research has been conducted throughout the world. This can be divided into two main groups:

Fuel management procedures

Engine management procedures

(1) Fuel Management Procedures

(a) Additives to Increase Cetane Number

Research by the VOLVO TRUCK CORP (14) has shown that the cetane number can be raised to 35 by the addition of 20% "cetanol" ($C_{18}H_{37}NO$) in methanol. This was sufficient to run the test engine under normal condition at 25°C ambient temperature, but it did not work at lower ambient temperatures and did not start. It was also found that additive could be reduced to 12% if the ambient air was preheated to 70°C. However, power output was reduced.

The latest ignition improver called "Avocet", which produces so-called "Diesanol" (originated from Diesel plus Methanol) when formulated in small amounts with methanol (1). Substitution of methanol for diesel results in a considerably modified torque curve response by comparison with diesel operation. These differences usually provide a more flexible driving response.

To date, however, this method has been considered to be impractical due to the high cost of additives.

(b) Emulsion of Alcohol with Diesel Fuel

(i) Chemical Mixing

The alcohol cannot simply be mixed with diesel. When an alcohol-diesel fuel blend is used, only part of the diesel fuel can be substituted with about 20 vol% of methanol and 15 vol% of solubilizers (15), good engine operation and advantage in full load performance as well as less smoke emission were obtained with the swirl-chamber engine and direct injection engine. Beyond the 20% limit, separation of the fuel in the injection system would lead to misfire or total loss of the combustion process. Also the cost of the solubilizer made the use of methanol/diesel emulsion economically unattractive.

(ii) Mechanical Mixing

Investigations by the Ontario Research Foundation (16) showed that an emulsion of 12% methanol (energy basis) was the maximum percentage possible if the engine manufacturer's rate of pressure rise was not to be exceeded when using a hydroshear emulsification unit.

Holmer et al developed a system to create the emulsion in the injection pump gallery by high pressure pulses and an irregular flow pattern (17). Test results reported that peaks of HC (including hydrocarbon and unburnt alcohol) above 1000 ppm were found to limit the methanol composition of the emulsion to 32% under full load conditions. This kind of system is still at a basic research level. In general, this method can only replace a small quantity of diesel fuel and gives unreliable starting conditions.

(2) Engine Management Procedures

(a) Dual Fuel System

Among the alternative fuelling options for CI engines, dual-fuelling is a well established technique. Two separate injection systems are used. A small amount of diesel is injected through a pilot injector to act as the ignition source for a larger quantity of alcohol fuel injected through the main injector at a slightly later time.

Berg et al (17) and Bindel (18) have identified the need for correct injection timing of the fuels since the pilot fuel must be injected prior to the injection of alcohol fuel. Early injection of the alcohol fuel resulted in misfire and a consequent increase in exhaust emissions.

Pilot fuel was required not only for ignition of alcohol, but also to ensure a smooth combustion process (19). In one of these dual fuel systems (20), up to 80% methanol was burnt with a brake thermal efficiency of 38.3%, which was 6.1% higher than that when only diesel fuel was used. All of the above systems showed brake efficiency for the dual fuel system was equal or greater than that for sole diesel operation, with significant reduction in the emissions of oxides of nitrogen. Other emission levels were reported nearly the same as the diesel.

This kind of system requires two complete sets of fuel handling equipment including injectors, injection pumps, fuel supply lines and fuel tanks. These significant costs discourage further development.

(b) Heated Surface and Glow Plug Ignition

This method aims to solve the difficulty of auto-ignition of alcohol fuel and can provide 100% replacement of diesel fuel.

It is important to locate the heated surface in such a position that enough fuel reaches it at all operation conditions (21). The brake thermal efficiencies at full engine output were higher than those obtained with diesel. But attention should be taken to avoid too much fuel contact with the heated surface as that could cause quenching and poor combustion.

Pischinger et al (22) developed an engine concept based on the ignition of the fuel-air mixture by a hot surface. This hot-surface ignition system (a diesel glow plug) was shown to display considerably improved exhaust gas emissions compared to the standard diesel engine. This was primarily due to the soot-free combustion of alcohols. The engine output power corresponded approximately to diesel engine operation and the efficiencies were almost the same in both cases. Variations in compression ratio, swirl intensity and the electric glow plug power effected engine load conditions.

Experience with the Deutz glow plug assisted methanol engine for transit bus operation (23) showed that it could fulfil the strict USA 1994 gaseous emission limit law without the need for exhaust gas after-treatment. However glow plug life and the lubrication oil consumption still need to be improved.

(c) Conversion of Diesel Engines to Spark Ignition

Various conversion systems of diesel engine to spark

ignition (24,25,26) have shown this to be an attractive conversion technique. The major concern is the high cost of the engine modification.

Spark-assisted diesel engines have the advantage of a lower compression ratio, smaller ignition power and less sensitive to spark plug position when compared to glow plug ignition. Seko et al (24) found a combustion system with locally concentrated spray, cuboid combustion chamber and low gas flow around the ignition plug could improve the brake thermal efficiency and achieve lower rate of pressure rise, while NOx emission and ignition stability were comparable to the sole diesel operation.

Three buses in which the engines had been converted to the spark-assisted ignition engine system were tested (26). The results revealed that it was possible to replace diesel in heavy duty commercial vehicles without loss of performance or operating range, but reliability and fuel economy would have to be improved.

(d) Exhaust Gas Throttling

A two-stroke alcohol engine has been developed to auto-ignite the alcohol fuel by controlling the residuals in the cylinder to raise the level of charge temperature (27). Thermal efficiencies were obtained close to those obtained with diesel operation. Significant advantages were gained in terms of a reduction in NOx, smoke and particulate emissions. The current problem concerns the high rate of wear of engine components.

(e) Fumigation.

Fumigation is a method in which alcohol fuel is

introduced into the inlet air by carburation or injection.

The Fumigation originated with dual-fuel systems and goes back to the early part of this century. Dr. Rudolph Diesel discussed this method of utilizing flammable gases in a diesel engine before 1900 in his British patent.

Havement et al (28) and Panchupakesan et al (29) added a carburettor to a diesel engine and used alcohol as the primary fuel. Small amounts of diesel or vegetable oil was injected through the conventional injection system to ignite the air-alcohol mixture.

Gary et al (30) used two pneumatic externally mixed atomizing nozzles to produced a stream of uniformly sized alcohol droplets (20 to 40 microns) in the intake air stream of an engine. The system allowed diesel engine operate on 40% to 60% ethanol with increased energy efficiency.

Murayama et al introduced ethanol up to 80% of the total energy supply into the intake manifold of a diesel engine without knocking (31).

Many other research (32, 33, 34) were also concerned with carburetting alcohol into diesel engines. Most previous studies were carried out focusing on thermal efficiencies and emissions. Kwon et al investigated the relationship between the ignition delay and ambient temperature (35). Methanol has been found to have a larger effect on the decreasing rate of ignition delay period than neat diesel fuel as a fumigation fuel. For the same temperature the ignition delay period decreased by increasing the fumigated fuel quantity into an inlet

manifold. This characteristic became larger at low air temperatures than at high air temperatures.

Alcohol fuels, especially methanol, has been studied at the University of Canterbury as an alternative fuel for transportation since the early 1980's. Completion of work on retrofit kits, which enabled petrol engines to operate on alcohol fuel (36), encouraged further investigation of alcohols operation on diesel engines. Another reason to carry out this type of research is that alcohols must be deployed as a substitute for diesel fuel, as well as petrol, if they are ever to be seriously considered as a substitute for petrol in order to eliminate any potential imbalance at the oil refining.

Early work by Green et al (37) described the development of a fumigation system that employed methanol not only as the fumigation fuel but also as the main body of injected fuel, thus obtaining total diesel substitution. This result followed a programme of work that involved operating a Ricardo E6 engine on pure methanol with high temperature heating of the inlet air supply. It was found that with a manifold air temperature greater than approximately 140°C degree under dynamometer conditions of no load normal engine operation existed.¹. However in the temperature range of 60°C to 140°C, a nearly 50% reduction in thermal efficiency occurred that was due to misfiring.

¹It was found that it was possible to run the Ricardo engine under the conditions of high power and high speed on methanol without the need for any additional air heating. But at low speed and low load the engine would not run without elevating the air temperature.

Combustion pressure analysis revealed a near linear relationship between increasing misfire and temperature decrease, see figure 2.1.

Since an engine misfire was always followed by normal combustion, a hypothesis was developed to explain the above phenomena:

Once a misfire had occurred, a small percentage of unburnt methanol and air become trapped in the clearance volume at the end of the exhaust stroke. This would become diluted during the following induction stroke, resulting in a very lean methanol-air mixture existing during the following compression stroke. At the end of compression just prior to the next fuel injection, there would be present in the cylinder a small percentage of fully vaporised methanol at an elevated temperature. It is this vapour, that would not normally be present, that formed a centre for ignition for the next charge of fuel.

To test this hypothesis, the engine was equipped with a small variable jet carburettor able to fumigate low level of methanol into the inlet air stream. Results, reproduced in figure 2.2, confirmed that it was possible to use a low level of methanol fumigation to enable the engine to operate without excess manifold air heating. This method forms the basis for the experimental work

presented in this thesis.

Table 2.1 summarizes various systems for diesel engines operation on methanol fuel. So far the techniques for 100% replacement of diesel have been at an applied study level and involve: spark ignition, heat surface and glow plug ignition. They seem to be feasible from a technical view point but engine modifications are rather complex and costly. The potential for fumigation seems to be attracted when considering its simplicity, therefore it is the focus for the work described in this thesis.

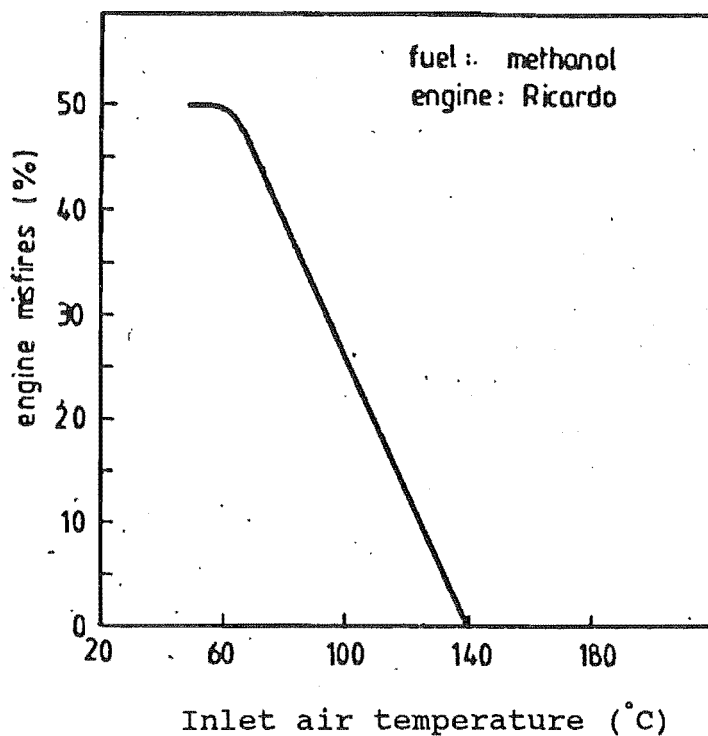


Figure 2.1 Effect of air temperature on engine misfire

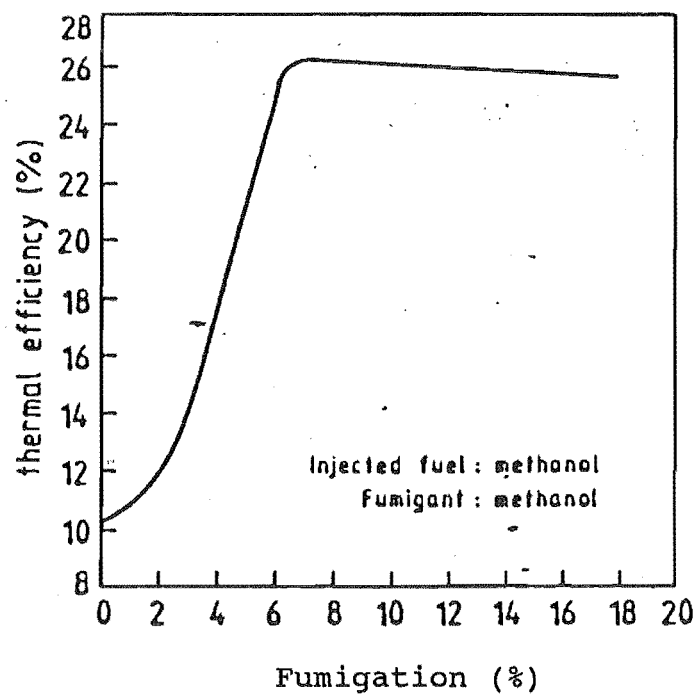


Figure 2.2 Effect of fumigation on thermal efficiency

Table 2.1 Features of Various Combustion Systems

Combustion system	Diesel-oil replacement (vol%)	Comparison with diesel engine					R&D status	Remarks
		Brake thermal efficiency	NOx	CO	HC	Aldehyde		
Dissolved fuel	20	Comparable	Comparable	Higher	Higher	---	Basic research	F. Pischinger DI
Emulsion fuel	20.8	Comparable	Higher	Comparable	Higher	---	Basic research	A. Lawson
Cetane number improved fuel	Unknown	Comparable	Lower	---	---	---	Applied study	J. Myburgh DI
Carbureted diesel	80	Lower	Comparable	Higher	Higher	Higher	Basic research	Chikahisa DI
Dual fuel injection	60-95	Comparable	Lower at high load	---	Higher at low load	---	Applied study	F. Pischinger DI
DI SI (I)	100	Lower a little	Lower	Lower	Lower	Higher	Applied study	T. Ullman Oxidizer
DI SI (II)	100	Higher	Higher	Higher	Higher	---	Basic research	Onishi No catalyst
Hot surface ignition	100	Lower a little	Half	---	Comparable	---	Applied study	C. Kroege
Residual gas ignition	100	Lower	Lower	Lower	Higher	Higher	Applied study	M. Jackson 2-cycle engine
Methanol gas engine	100	Lower a little	Lower	Higher	Lower	---	Applied study	H. Bergman

CHAPTER III

THEORETICAL CONSIDERATIONS

I. INTRODUCTION TO COMPRESSION IGNITION COMBUSTION

Before a study of combustion changes caused by fumigation some fundamental aspects of diesel combustion theory should be addressed.

(1) Compression Ignition (38,39)

The CI engine operates with a heterogeneous charge of previously compressed air and a finely divided spray of liquid fuel. Fuel injection is initiated when the air that is compressed in the cylinder reaches temperatures in the range 800K to 900K and at a pressure ranging from 30 to 35 bar. Injection lasts approximately 20 to 30 degree of crank angle, and the injection rate is approximately 6 to 10 mm³ of fuel per degree of crank angle. The combustion chamber geometry and injection characteristics are therefore optimised for the above process.

Through the study of cylinder pressure diagrams and the rates of heat release, the combustion process can be divided into four stages. These are illustrated in figure 3.1 (39).

(i) Ignition Delay Period - This lasts from the initiation of injection to the instant of appearance of a hot flame (self-ignition), which is accompanied by a rise in pressure-time curve. This period consists of the physical and chemical ignition delay, the heat of the

compressed air is transferred to the sprayed fuel, which begins to vaporize and diffuse away from the core of the injection jet. Thus heat release rate would be negative on the diagram of heat release versus crank angle.

(ii) Period of Rapid Combustion - This starts from the point of ignition and is a period of uncontrolled combustion. The pressure rise proceeds at an increasing rate and continues until most of the fuel already injected has been burnt. The rate of heat release reaches its maximum. At the end of this period, the rate of heat release decreases and the combustion processes ceases to be controlled by chemical kinetics.

(iii) Period of Burning at Quasi-Constant Pressure - In this period, combustion continues at a rate determined by how quickly the remaining fuel is injected into the combustion chamber and finds oxygen necessary for combustion, hence the injection process can have a marked effect on this period of combustion. As burning spreads during this third period to fuel which may be only partially evaporated, free carbon is generated and causes the flame to become luminous.

(iv) Period of Fuel after Burning - Any remaining droplets are finally burnt and oxygen diffusion continues to control the burning process. This final burning overlaps with the expansion stroke and even sometimes with the exhaust stroke, thus engine efficiency drops as a result of incomplete and imperfect burning. The temperature in the cylinder reduces due to the expanding volume and the remaining unburnt mixture burns at a much

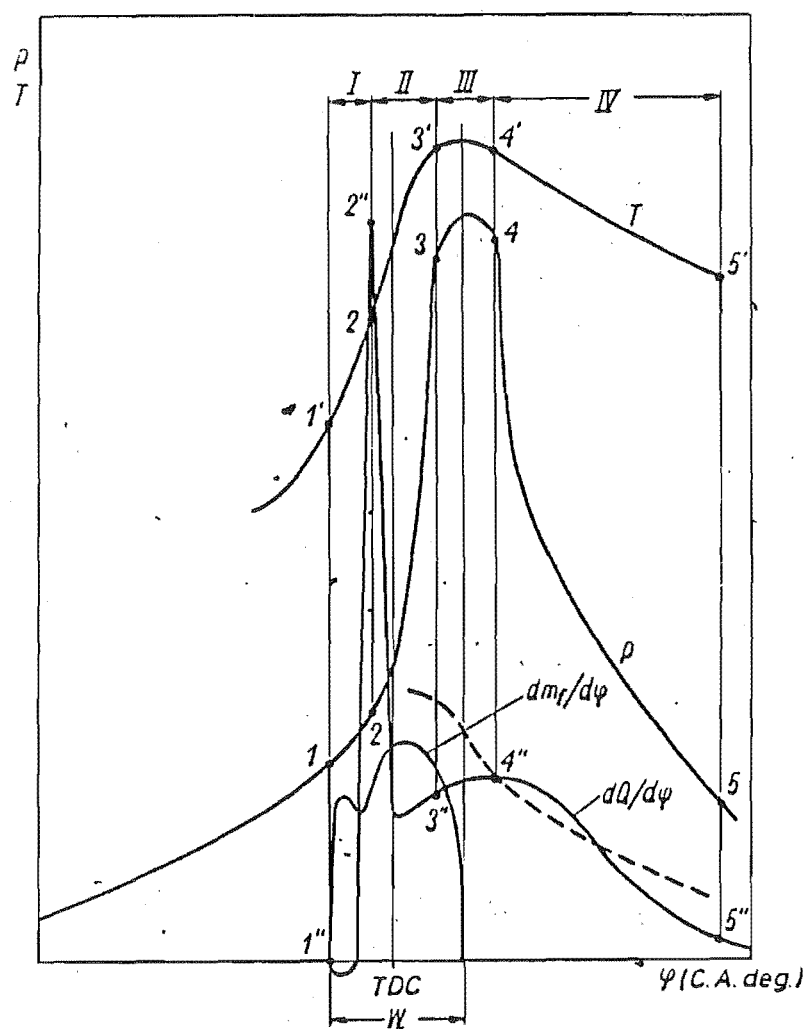


Figure 3.1 Diagram of pressure, temperature, heat release $dQ/d\psi$ and injection rate $dm_f/d\psi$ versus crank-angle in diesel engine

lower rate.

As the ignition delay period increases, more fuel will enter the combustion chamber and the air temperature will be higher at the end of this period. Hence there is a quicker and greater rise of the uncontrolled kind of rapid combustion when ignition finally does occur. The most noticeable effect of a long delay period is increased engine noise. That is the so-called "diesel knock", which causes rougher running particularly at lighter load. This phenomenon would apply to systems using fuels with high latent heats of vaporisation, resulting in the problem of long ignition delay periods.

(2) Fumigation Effects on Combustion Process

The alcohols have a poor self-ignition capacity because of their low cetane ratings and large latent heat of vaporisation. Fumigation methods can be used to ensure positive ignition but the problem of long ignition delay and homogeneous combustion still arise. In this research, the alcohol fuel is mixed with the intake air in a small jetted carburettor. This type of combustion is typical of spark ignition engines, where the fuel and air are mixed prior to compression in the cylinder with fumigation. Since there is a certain ratio of fuel/air existing in the cylinder before injection, combustion occurs within a range of fuel/air ratio. Under very lean condition, misfire could occur during light loads because of an insufficient temperature for ignition. On the other hand, during combustion, knocking could occur because of too long ignition delay caused by the high latent heat of

vaporisation and the high auto-ignition temperature of alcohol fuels.

For the normal homogeneous combustion process, a flame is initiated from a controlled ignition source and it is supposed to propagate steadily throughout the rapid combustion period. The fuel/air contents are assumed to be homogenously mixed. But there are some factors such as variations in mixture strength, turbulence and property gradients that result in a certain degree of variability within the combustion process, this is called cyclic variation and results in fluctuations in the combustion pressure curve, these are typically more pronounced for spark ignition than compression ignition engines. In the fumigation combustion process, that is similar to spark ignition engine, there are also variations in mixture strength, that will also tend to increase cycle variability.

With the practical use of the fumigation method, careful control should be taken to minimise the cycle-to-cycle fluctuations in order to gain smoother engine output and prevent excess stress in engine components.

II. FUEL CHARACTERISTICS

When the diesel engine runs on alcohol fuel, the performance of the engine and exhaust emissions will be different from that when operating on diesel. This is due to the different physical and chemical properties of alcohol fuels when compared to diesel fuel. Diesel engines have been developed for over century and are designed to

give their optimal performance when running on diesel. In this study, only the minimum modification was made to the engine to achieve the target of a low cost conversion. In this case, fuel characteristics will play an important role on the combustion process. This section discuss the important characteristics of diesel and alcohol fuel and their effects on combustion.

Diesel fuels chemical composition depends mainly on the composition of cruel oil and subsequent refinery treatment, whilst the alcohols have a simple constant molecule structure. Table 3.1 lists the important properties of alcohols and diesel fuel.

(1) Diesel Fuel Properties

(a) Diesel Composition (38)

Diesel fuel is a mixture of different hydrocarbons which has its boiling point in the range of 180°C to 360°C. It is not practical to identify all the constituent parts for a particular crude. However they fall into three main series known as paraffins, naphthenes and aromatic compounds. Amongst these, paraffin molecules are most easily attached to oxygen to form compounds with low ignition temperatures. Diesel has a low auto-ignition temperature 250°C and flash point of 55°C. This is good enough to auto-ignite the diesel fuel when it is injected into the compressed hot cylinder air, even at a compression ratio as low as 12:1.

(b) Ignition Quality

One of two basic requirements for ignition and

	Methanol	Ethanol	Diesel
<u>CHEMICAL PROPERTIES</u>			
Formula	CH ₃ OH	C ₂ H ₅ OH	C _n H _{1.86n}
Molecular weight (g/mol)	32	46	13.86n
Main constituents by weight:			
% carbon	38	52	86
% hydrogen	12	13	13
% Oxygen	50	35	-
% Sulphur	-	-	0.8
<u>PHYSICAL PROPERTIES</u>			
Density (kg/l)	0.79	0.79	0.84
Viscosity at 20 °C mm ² /s	4	1.5	0.75
Boiling temperature (°C)	65	78	180-360
<u>COMBUSTION PROPERTIES</u>			
Stoichiometric Air/Fuel ratio (kg/kg)	6.5	9	14.5
Lower calorific value (MJ/kg)	19.7	26.8	42.5
Energy content related to diesel (v/v)	.464	.632	1.0
Specific latent heat of vaporisation (kJ/kg)	1110	904	250
<u>IGNITION PROPERTIES</u>			
Autoignition temperature	450°C	420°C	250°C
Cetane number	3	8	50-55
Octane number (RON)	106-108	106-108	-
Ignition limit % by vol. of gas in air			
Lower	5.5	3.5	0.6
Upper	26	15	6.5

Table 3.1. Properties of the alcohols and diesel fuels

subsequent combustion is adequate energy source, which causes self ignition of fuel injected. Hence the compression temperature and pressure must be sufficiently high to enable the engine to start from cold and when running to give adequately smooth combustion. It requires efficient combustion chamber design as well as the correct chemical characteristics of the fuel.

When the fuel is injected into the cylinder near to the TDC position, if the fuel doesn't burn almost instantaneously, then considerable amount of fuel will accumulate, This will cause a sudden rapid pressure rise and knock badly when combustion finally starts. This will cause the engine components to be subjected to severe mechanical stress and lead to the premature failure of components.

The cetane number indicates the ignition quality of diesel fuel. The higher the cetane number is, the more easily the fuel can be ignited. Diesel fuel usually has a cetane rating of about 50. This ensures good auto-ignition of diesel fuel as well as smooth operation.

(c) Air/Fuel Ratio

The ideal air/fuel ratio for complete fuel combustion is called the stoichiometric ratio, which is rated at 14.5:1 for diesel, and is an important factor in ignition and subsequent combustion. Compared to spark ignition engine, diesel engines can only use up to about 90% of all the air induced. But diesel engine can employ full air admission at reduced loads. The air/fuel ratio might be as high as 100:1 at idling speed. This ability of the diesel engine to use air admission results in a gain over the throttled spark engine as load are reduced.

In general, the overall air/fuel ratio for a diesel engine is always weaker than the stoichiometric ratio.

(d) Other Diesel Properties

Compared to the chemical characteristics, the physical characteristics are less important. It has been generally held that volatility and viscosity do play a small part through differences in rate of evaporation and droplet size respectively, but research has shown that with a wide range of distillate fuels there is very little difference in fuel spray penetration (40). In engine tests quite significant differences in the boiling range of fuels displayed little effect on power, fuel consumption and smoke (41).

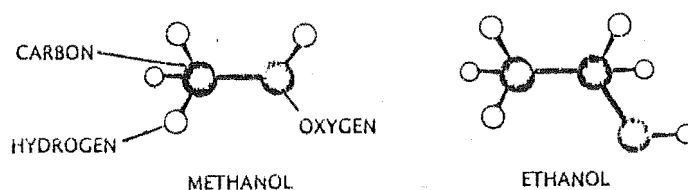
Diesel fuel still plays some part in the deposit and wear rate problems of engines. Poor fuel handling and poor injector maintenance are the main reasons for poor atomization, injection nozzle dribbling, carbon deposits and even orifice blockage. Products of combustion that include water vapour, carbon dioxide and sulphur oxides (from the sulphur content of the fuel) contribute to the corrosion, which leads to wear problems. Abrasive wear is associated with a breakdown of lubrication.

(2) Alcohol Fuels Properties

(a) Alcohol Composition

Methanol and ethanol are the two important varieties of alcohol. And this research is involved with these two promising fuels. The main factor which causes the major differences between the properties of alcohols compared with conventional hydrocarbon fuels is the molecule structure. Methanol (CH_3OH) may be considered as two molecules of hydrogen

($2H_2$) chemically liquified by a molecule of carbon monoxide (CO), while the ethanol is just a paraffin group (CH_2) added to methanol. Therefore they share very similar properties. In the case of methanol, oxygen constitutes 50% of its molecule by weight which makes it strongly polar especially when compared to the non-polar hydrocarbon fuels.



Because of the simple structure of alcohol fuels, they burn cleanly, forming mostly carbon dioxide and water. On the other hand diesel fuel is complex, variable and contains many bonds between carbon atoms. Such molecules are more likely to leave unburnt and photochemically active hydrocarbon compounds.

(b) Latent Heat of Vaporisation

The polarity is responsible for a strong hydrogen bonding and the high latent heat of vaporisation. This gives methanol a specific latent heat of vaporisation of 1100 kJ/kg which is 4.44 times as large as that of diesel fuel at about 250 kJ/kg. When a fumigation method is adapted to the diesel engine, the alcohol/air mixture requires preheating because of this large value of latent heat. The most practical way is to preheat the intake air so as to avoid condensation of alcohol that would occur if the air heating was applied after mixing.

(c) Calorific Value

Although alcohols have higher energy content per volume than other alternative fuels, they still have low energy contents when comparing with an equal volume of diesel.

Ethanol energy content is 63% and methanol just 46.4%. This requires about 5.7 and 9.6 times as much heat to evaporate sufficient ethanol and methanol fuel to deliver the same heating value as diesel fuel. When alcohol is used in diesel engines, the fuel pump should be capable of producing higher fuel flows to compensate for the low energy content of alcohols. Exhaust heat could be recovered to improve the vaporisation of alcohol thus raising the overall thermal efficiency.

(d) Air/Fuel Ratio (12)

Compared to the stoichiometric air/fuel ratio for diesel, alcohol fuels have a far lower value, the methanol ratio is about 6.5:1 while ethanol is 9:1 (mass basis).



From table 3.2, the air/fuel ratio for the alcohol fuels is seen to be lower since oxygen constitutes a high percentage of the weight of the alcohol molecule.

Fuel	CH ₃ OH				C ₂ H ₅ OH			
	O ₂		Air		O ₂		Air	
	moles	kg	moles	kg	moles	kg	moles	kg
Fuel	1.0	1.000	1.0	1.000	1.0	1.000	1.0	1.000
Oxidant	1.5	1.498	7.295	6.449	3.0	2.085	14.589	9.113
Reactant	2.5	2.498	8.295	7.549	4.0	3.085	15.589	10.113
Products	3.0	2.498	8.795	7.549	5.0	3.085	16.589	10.113
% Increase	20.0		6.0		25		6.41	

Table 3.2 Stoichiometric oxidation quantities

(e) Ignitibility

The ignition limit ranges of alcohol fuels are

greatly extended, especially at the upper limit, when compared to diesel. Low energy content and high latent heat of vaporisation requires more fuel to obtain the equivalent power output as diesel. The alcohols have a high octane number which indicates high resistant to knock or spontaneous ignition when they are used in spark ignition engine, but this is opposite to the spontaneous compression ignition required of a diesel engine fuel.

Alcohols have such low cetane numbers that they cannot be measured directly. Hence they are not really suitable for use in a conventional diesel engine. However they can be used in conjunction with another fuel that has good compression ignition qualities or when modifications to the engine have been carried out to promote ignition.

Another measure of ignitibility is the lowest temperature at which the air/fuel mixture ignites spontaneously. This depends on engine operating conditions and for the alcohols range from 400°C to 500°C . Addition of water will slightly increase the ignition temperature. This temperature is however still below that of the air compressed in the cylinder at the end of compression stroke, thus they can still be considered for use in a diesel engine.

Methanol is known to have a correspondingly long ignition delay. The ignition delay period of a fuel is a very complicated process, which can be split into two main periods. It consists of physical and chemical delay periods as shown in figure 3.2.

The vaporisation of alcohol results in the cooling

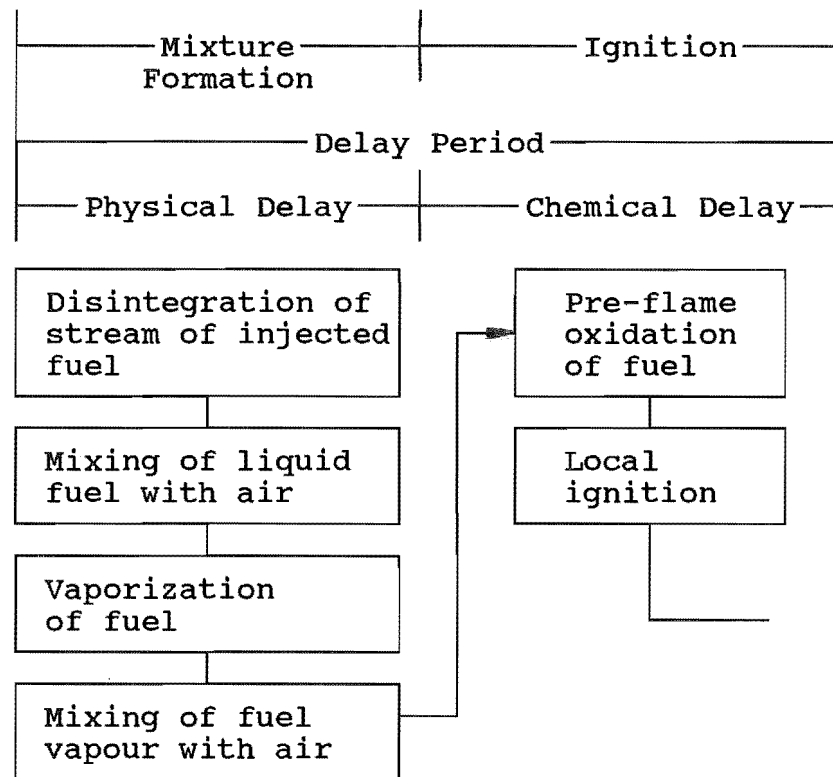


Figure 3.2 Ignition delay period

of the air and the combustion charger, and increases the length of the physical delay period of the fuel. Foster et al (42) concluded that an increase in the percentage of alcohol will result in lower temperatures and pressures at the start of injection which would tend to increase the ignition delay period and cause a slightly lower flame temperature. Recent research (35) has found the total ignition delay period to have little difference when a diesel engine was partially run on alcohol by fumigation when compared to that with diesel fumigation.

Kwon et al (35) noticed that increasing the quantity of methanol to the inlet manifold reduced the compression temperature because of the heat that was required to

evaporate the methanol. On the other hand, formation of the pre-mixture was improved by adding methanol to the intake air. Low temperature oxidation reactions of methanol-air mixture occurred during the compression stroke and intermediate compounds were created with heat release in the cylinder, which caused ambient conditions in the combustion chamber to be suitable for easy ignition. This mixture then ignited the main injected fuel. Thus the ignition delay was decreased by increasing the methanol quantity under the same air temperature, this is most clearly demonstrated at the lower temperature of 700K, figure 3.3.

Another issue concerned with fuel ignitibility is cold start. The cetane numbers of the alcohols are too low to achieve auto-ignition, especially at the cold start. Certain programmes to auto-ignite low cetane fuel have mainly utilized higher compression ratios or inlet air heating in order to raise the compression temperatures to levels at which these alternative fuel will ignite (43, 44). They have not met with consistent success because of structural limitations such as peak cylinder pressure and engine durability problems.

Detroit Diesel Allison Diesel (DDAD) have developed a two-stroke cycle which has suitable characteristics that make compression ignition possible (27). The inherent characteristics of this engine have been exploited by controlling the scavenging process to produce the desired in-cylinder conditions at the time of fuel injection. However, for the ignition of ethanol it was found that cold

starting, even with a compression ratio 20:1, was impossible without glow plugs or preheating the engine. A dual fuel system with switch-over was suggested as a way to start and warm up the engine on diesel.

The literature concerned with pure alcohol use in diesel engine without ignition aid is rare. Research by Foster et al (42) concerned with high percentage of alcohol fumigation still used a small amount of diesel as the pilot fuel to ignite the pre-mixture of air/alcohol, this also avoided encountering the problem of cold start.

(f) Knocking and Misfire

It is a common knowledge that alcohol fuels cause misfire at low speed and reduced load and knocking always appears when an excess amount of alcohol is used to replace diesel.

Murayama et al (31) found that the combustion chamber configuration has a big effects on the level of alcohol substitution. A decrease in pre-chamber to clearance volume ratio reduced the diesel knock and eliminated misfire at low compression ratios, permitting a significant increase in the amount of ethanol introduction. Figure 3.4 and 3.5 show this change.

Foster et al (42) used ethanol substitution of up to 80 percent of total energy in a turbo charge diesel engine. Vaporised alcohol increased the substitution percentage and the engine still run smoothly at reduced load. Anti-knocking limits could be raised when vaporizing the alcohol. The disadvantage of this maximum level substitution of alcohol was the excessive hydrocarbon

emissions (>7000 ppm).

(g) Combustion Pressure

Knocking and misfire will occur when fumigating alcohol. Knocking combustion, accompanied by an increased level of peak pressure, will increase the stress and wear on engine components.

The DDAD two stroke compression ignition engine was expected to meet auto-ignition criteria and have stable operation. Thermal efficiency were comparable to diesel engine operation but peak cylinder pressures at rated conditions were above the engine design level (27).

An optimal fumigation concept developed in the CUMINS Engine Company for partial substitution of alcohol in turbo-charge DI engines revealed a lower rate of pressure rise and lower peak pressure. However there were two initial heat-release peaks. The shape of the first peak is similar to that of diesel combustion only. The second larger premixed peak is probably due to the alcohol charge burning subsequent to the initial diesel combustion (32).

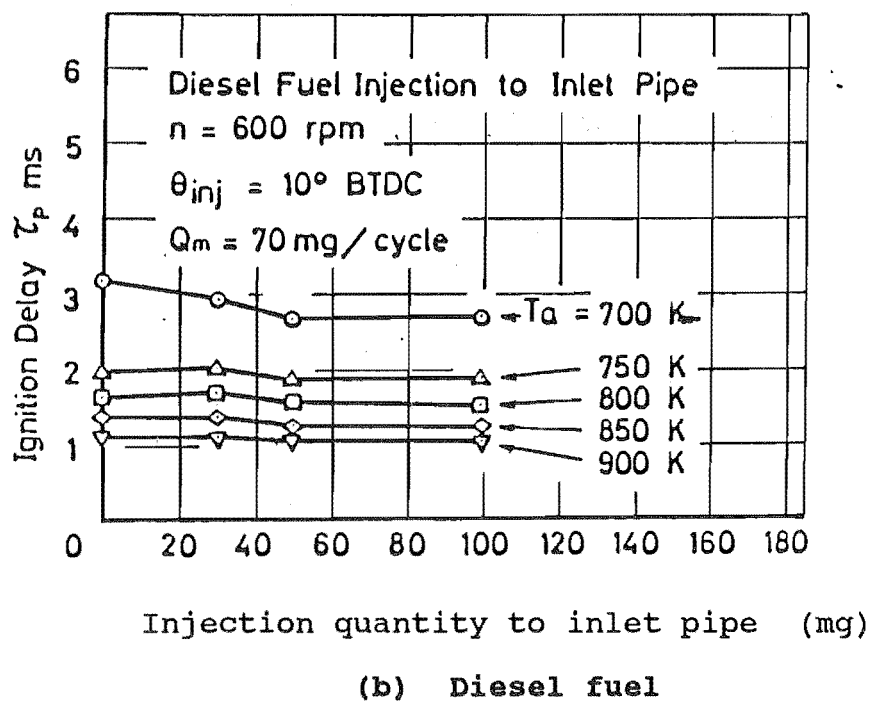
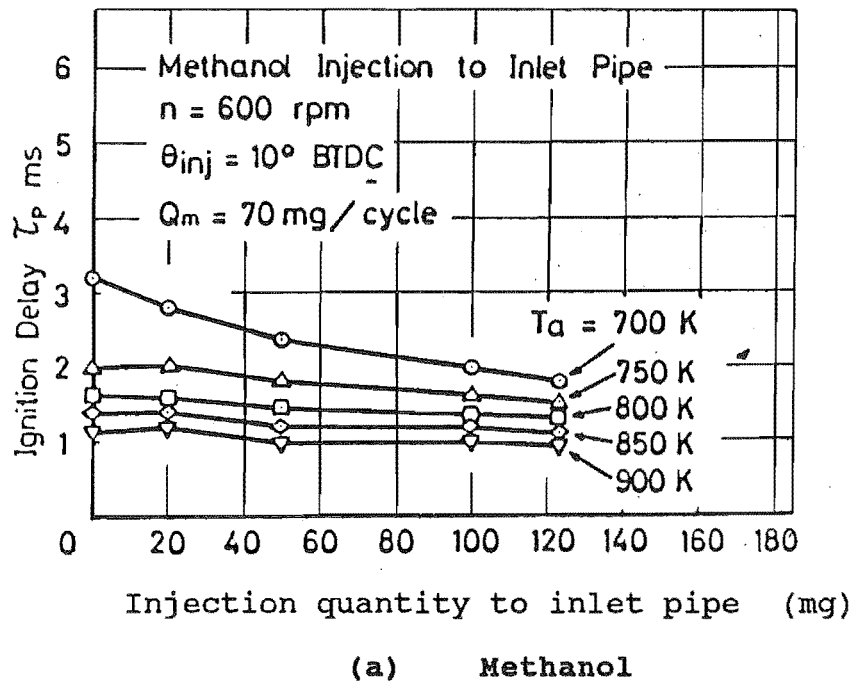


Figure 3.3 Relationship between ignition delay and fumigated fuel quantity to inlet pipe

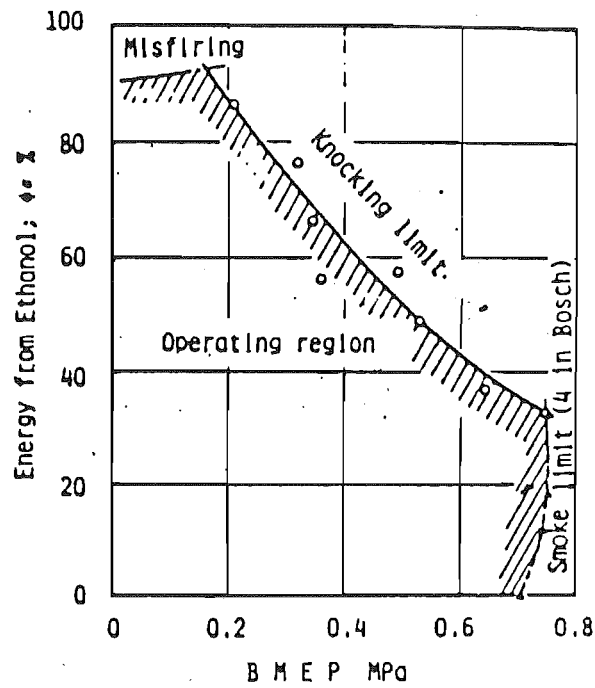


Figure 3.4 Operation region limited by smoke, knocking and misfire in a standard diesel engine

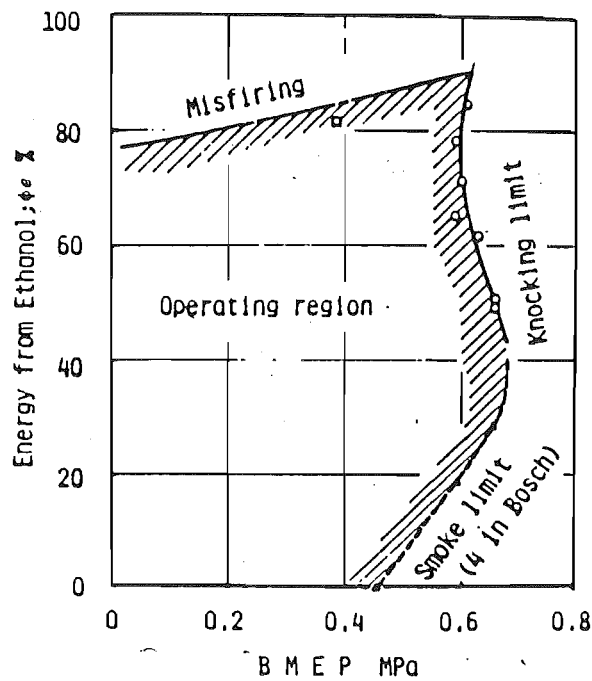


Figure 3.5 Operation region limited by smoke, knocking and misfire in a modified diesel engine

CHAPTER IV

CONVERSION OF DIESEL ENGINE TO ALCOHOL FUELS

I. INTRODUCTION

Experimentation was carried out using a single cylinder 507 cc Ricardo E6 variable compression engine, which was fitted with a "comet" cylinder head for diesel operation. The engine and associated dynamometer were fully instrumented to record engine performance data. Table 4.1 details the E6 test engine.

Type	4 stroke, indirect injection
Aspiration	natural
Cylinder	single
Bore * Stroke	3 inch * 4 3/8 inch
Capacity	507 cc
Compression ratio	6 to 20.0
Max. power	about 5 kW
Engine speed	range 750 to 2500 rpm
Others	Variable injection timing (30° - 45° BTDC) Inlet throttle (0-100%)

Table 4.1. Ricardo E6 variable compression engine

At a later stage in this study, a data acquisition system was built up to integrate the measurement of engine performance parameters with control of the level of fuel fumigation and the power input to an additional manifold heater, which was used to control the temperature of inlet air. These will be discussed later.

II. ENGINE MODIFICATION

The main modification to the diesel engine was the additional of a jet carburettor, which was mounted on the inlet manifold. The inlet air was mixed with alcohol inside the carburettor, see plate 4.1. This is carburation or fumigation. The fumigation level is controlled through movement of a shaft with cone end inside. The outside end is fixed to an indicator arm which points to different settings on a round disk for indication purpose.

As mentioned earlier, alcohols have a disadvantage with regard to their high latent heat of vaporisation, and preheating the intake air will improve the vaporisation of the fumigated alcohol in the carburettor and inlet manifold. This will improve part load fumigation performance. A hotter charge is beneficial for the vaporisation of alcohol fuel and in reaching a more homogeneous mixture. It also increases flame speed, improves ignitability and reduces wall quenching. The chemical and physical processes occurring during the ignition delay period are speeded up, resulting in earlier combustion and higher peak temperatures. Since the hotter intake air is less dense, the air component of the mixture is effectively reduced. This leads to a richer charge being induced into the cylinder with a consequent improvement to efficiency.

A 1.0 kilowatt electric heater was sited up stream of carburettor, which could provide a variable energy supply to the inlet air in order to obtain a required air temperature. This enhanced the power provided by an

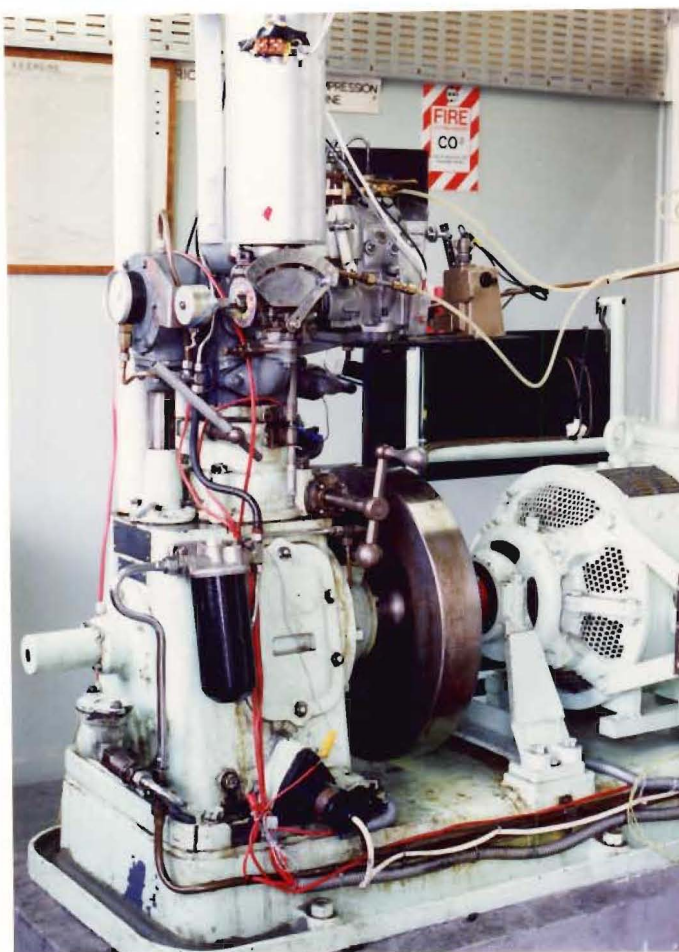


Plate 4.1 Overall lay-out of the E6 Ricardo engine

existing 500 Watt heater sited inside the air flow meter, provided by the Engine Manufacturer.

III. TEST EQUIPMENT

(1) Overall Lay-out of Test Cell

The E6 variable compression engine is situated in test cell #2 of Thermodynamic Laboratory at the University of Canterbury. It is a durable Ricardo engine with a comet Mark III pre-combustion chamber. This engine was designed to suit a variety of research purposes. The test cell is semi-soundproofed and a glass window allows the engine to be visually monitored from outside the room. Exhaust fumes are drawn down an exhaust pipe and into a main chimney. There is an air-conditioning unit and a fan on the ceiling for heating and ventilating of the test cell.

(2) Engine Torque

The E6 variable compression engine is coupled to a D.C. dynamometer. This permits direct motoring and the electric power output from the dynamometer is dissipated in a resistance load bank of negligible temperature coefficient. The dynamometer's casing is mounted in trunnion bearings and torque is indicated by the force at the end of a radial arm. The force can be measured by the balancing of a spring balance. The total mass of weight minus the scale reading of the spring balance times the torque arm distance gives a reading of the engine torque.

(3) Engine Speed

Engine speed is monitored both by a mechanical

tachometer driven by belt and an analogue meter, which takes a signal from a pulse generator sited close to a ring gear.

To minimize speed fluctuations whilst operating the engine on alcohol fuels, Bosch AGB-130 auto-speed control unit was integrated with the control unit of the dynamometer.

The principle of operation of the auto-speed system is to compare the frequency of a signal generated by the engine, with a signal generated by the control unit. The signal is processed by the solenoid which controls the rack setting of the injection pump to maintain equal signals, i.e. constant speed.

The signal generated by the engine is picked up by a magnetic unit mounted over the ring gear of the engine. Previously such signals were produced from self-tapping screws mounted on a flexible coupling between the engine and dynamometer. This was replaced by a ring gear because the screws were not well mounted and caused fluctuations of signal which resulted in poor speed control.

(4) Combustion Pressure Measurement

A piezo electric pressure transducer model AVL-8QP 500C was used to measure the engine cycle pressure.

The signal from a pressure transducer is so small that it requires amplification before a pressure versus crank angle trace can be monitored. The pressure transducer signal was amplified by a **CUSSONS** P-V channel that provided a suitable trace on an oscilloscope.

The pressure transducer calibration is referred in

Appendix A.

(5) Fuel Consumption**(a) Diesel Fuel**

The volume of fuel consumed over a given time period was measured using a glass column with reading accuracy to ± 0.1 ml. The fuel flow rate may be recorded manually or by use of a fuel flow transducer made by PIERBURG (type 116-H), which outputs both the digital and analogue signal.

(b) Alcohol Fuels

This was similar to the diesel fuel measurement, but the alcohol fuels were measured by means of two glass columns. One was used to measure the main fuel flow to the injector, whilst the other measured the fumigation flow rate. The fuel flow was recorded for a period of two minutes or for the maximum volume of fuel in the glass column, whichever was the shorter. The accuracy for the fuel consumed was ± 0.1 ml, and ± 0.1 second for the time over which the fuel was consumed.

The PIERBURG fuel flow-rate transducer was mounted to monitor the total fuel consumption, and was used to collect data to determine the thermal efficiency of the engine.

(6) Air Flow Rate

The ALCOCK viscous air flow meter produced a pressure differential which was measured by means of a manometer. The reading was recorded during testing and later processed by a computer program to calculate the air flow rate.

(7) Temperature Measurement

Unshielded chromel-alumel (Type K) thermocouples were used to measure a variety of engine temperatures. The thermocouple emfs were monitored by a digital reading voltmeter on the **CUSSONS** control unit. The following temperatures were generally recorded:

Ambient air temperature

Inlet temperature (air temperature after being heated)

Manifold temperature (mixture of alcohol & heated air)

Exhaust Temperature

Cooling water temperature

Oil temperature

CHAPTER V

BASELINE TEST RESULTS AND DISCUSSION

I. INTRODUCTION

The aim of this study was to design, build and test a fumigation and control system that could be adapted for automotive use. All of the necessary data was collected with the engine running on diesel and alcohol fuels and is presented in this chapter. Meaningful parameters were calculated using computer programs and appropriate graphical displays are presented, The information contained in these displays is essential for the development of the control system.

The engine test consisted of three phases. The first phase was to collect engine performance data when operating on diesel, These results were used as a reference for comparison with alcohol fuels. The second phase was to perform various tests with the alcohol fuels using the fumigation method, different engine set-ups were examined to investigate their effect on performance. In the third phase carefully controlled experiments were used to decide the best engine set-up such as fumigation level, manifold temperature etc. Analysis of the above results led to the development of a control system based on an analogue/digital conversion system. A computer program was

written to achieve the practical control of the Ricardo E6 research engine. The control procedure will be described later.

(1) Fuel Preparation

In diesel engines, diesel fuel is used not only as the energy source but also for the lubrication of the fuel pump and injection components. Alcohol fuels have a very high viscosity value that could cause the fuel pump lubricity problems. Another problem associated with pure alcohol fuels is corrosion of engine components.

To solve these problems, two percent of water was added to the pure alcohol fuel to reduce its corrosive effect (45). Then two percent of castor oil (unburnt) was mixed with the above alcohol-water mixture to ensure adequate lubrication. The castor oil must be mixed well by shaking to ensure correct mixing.

(2) Starting

It is well known that diesel engines are difficult to operate on alcohol fuels at low speed and reduced load conditions, and that cold start is a further problem. A simple switch-over unit was built to overcome this problem. Diesel fuel was used to start the engine until it was warmed prior to switching to the alcohol fuel supply.

II. EQUATION (46)

Engine power:

$$B_p = T \cdot 2 \cdot 3.1416 \cdot N / 60 \quad (W) = (9.81 \cdot W - S) \cdot N / 20895 \quad (kW)$$

$$\text{Air flow-rate:} \quad V_a = 0.02104 \cdot K \cdot h \quad (m^3/hr)$$

$$M_a = \frac{P \cdot V_a}{R \cdot T} = \frac{0.5863 \cdot K \cdot h \cdot P_{atmos}}{273 + T} \text{ (kg/hr)}$$

$$\text{Fuel consumption: } M_f = \text{Vol}_f \cdot 3.6 \cdot \text{SG} / t_f \quad (\text{kg/hr})$$

$$\text{FC} = (\text{FC})_{inj} + (\text{FC})_{fumi} \quad (\text{kg/hr})$$

Brake thermal efficiency:

$$\eta = \frac{B_p \cdot 3600}{\text{FC} \cdot \text{LCV}} \cdot 100\%$$

Equivalence ratio:

$$\lambda = \frac{(M_a/M_f)_{actual}}{(M_a/M_f)_{stoichiometric}}$$

Indicated mean effective pressure:

$$\text{Imep} = \frac{\oint p dv}{V_{swept}}$$

III. ENGINE PERFORMANCE

One of the most important indications of engine performance is its thermal efficiency. It is defined as the fraction of fuel energy supplied to the engine which is converted into useful power, hence it takes into account all of the losses involved in the transfer of energy from the fuel to the final useful power output to the drive shaft. Thus it is a useful parameter to compare the performance of different fuels. Alcohol fuels have about half the lower caloric value of diesel. Therefore absolute alcohol fuel consumption will be approximately double when

used in diesel engines. This makes comparison between alcohol and diesel fuel impossible without referee to thermal efficiency or specific fuel consumption.

(1) Diesel Operation

Figure 5.1 displays baseline performance data for diesel engine operation over a range of engine speeds. These results were obtained at an engine compression ratio setting of 20:1 and a standard injection timing of 35°BTDC. These curves are to be used as reference data for the following alcohol tests.

(2) Alcohol Operation

The lower calorific value of methanol is lower than that of diesel. In order to obtain the same engine power as diesel fuel, it is necessary to operate with approximately 2.1 times more methanol (mass basis). The Ricardo injector pump was able to supply substantially more fuel than is required for the diesel fuel operation, but not sufficient to supply the desired flow rate of methanol. During testing, the rack setting of the governor was at its maximum position, but power output was still far short of that with diesel. Increasing the engine load would cause a drop in engine speed because of insufficient fuel supply.

The maximum power output with methanol fuel could reach the level of diesel, but only with a high level of fumigated methanol mixed with heated inlet air. Figure 5.2 shows the variation of brake thermal efficiency for the engine when operating totally on methanol with fumigation. The power outputs and thermal efficiency are similar to

diesel due to a combination of methanol injection and methanol fumigation. This meant a high energy input to the inlet air to improve the engine performance, and this high level of fumigated methanol could compensate the inadequate energy content of the injected methanol.

In order to achieve a lower level of fumigation and associated lower heating to the inlet air, the fuel pump was modified to obtain a higher flow rate. The first trial was to increase the injector pump supply by approximately 100%, which corresponded to the fact that the theoretical energy content of methanol to diesel was about 50%. Testing showed the engine would not operate normally with this modification, no matter what the fumigation level or intake air heating level. Too much fuel was injected into the cylinder which quenched the combustion process and caused a misfire under reduced load. Under high load and high speed conditions, the engine could maintain operation for a few moments before serious knocking occurred due to the long ignition delay period caused mainly by the physical delay process. Cylinder head gaskets were broken twice because of the high peak pressure caused by knocking, see plate 5.1.

In contrast, the fumigation level could be as high as 80% under reduced load conditions without encountering knocking, and the actual methanol consumption was less than the theoretical requirement hence increasing thermal efficiency. The injection pump was therefore modified to supply 20% more fuel to meet the performance requirements.

Figures 5.3 and 5.4 show the engine performance when

operating on methanol and ethanol after modification of the injector pump. Two sets of test results are shown, i.e. minimum and maximum levels of fumigation, over a range of engine speeds. Minimum fumigation refers to a fumigation level just sufficient to avoid misfire whilst the maximum level is determined by the onset of knocking.

Engine speed was maintained by an auto-speed unit which controlled the rack setting of the fuel governor to compensate for a load change. When there was a change in fumigation level, the fuel flow to the injection pump would be altered accordingly, thus maintaining a constant speed.

Throughout the range between maximum and minimum fumigation levels, the engine ran entirely on the alcohol fuels. The thermal efficiency varies, and this was dependant on the level of fumigation. It is clear that thermal efficiency is greatly improved by increasing the level of fumigation at a set power output, e.g, methanol can reach a maximum thermal efficiency of 25.5% with a maximum fumigation level at 1500 rpm whilst the highest thermal efficiency for minimum fumigation level is less than 20%.

Figure 5.3 shows that two curves intersect for resulting at 2000 rpm. This was because the rack was at its maximum position to obtain the maximum power output and in this situation fumigation only adds to the output power instead of improving the thermal efficiency.

Ethanol as a fuel displays a similar performance to those obtained for methanol although the thermal efficiency is reduced at the highest power settings of 2000 rpm. This

is a direct consequence of the lower levels of fumigation that could be employed without knocking.

IV. FUMIGATION LEVEL

From the literature, fumigation has attracted wide interest as it requires only minor changes to the engine instead of certain expensive modifications that can be undertaken. However, the maximum substitution limit for diesel has been reported as 80% of total energy supplied (31). In many other cases, high fumigation levels were limited by occurrence of knocking, which can cause premature shortening of engine life.

The fumigation levels displayed in figures 5.5 and 5.6 correspond to the engine performance discussed previously.

(1) Minimum Fumigation Limit

For most of the operating conditions, there is a minimum fumigation level below which combustion becomes intermittent and the engine is unable to maintain speed and load, this condition is deemed to be unstable. At the higher speed and load conditions, the engine could run satisfactorily on the alcohol fuels without fumigation. This is in agreement with the literature (37,47). Fumigation levels increase significantly to about 40% at 1000 rpm. The intake air must also be heated to obtain a favourable temperature to assist the vaporisation of the fumigated alcohol. Ethanol had lower fumigation levels under reduced load due to its higher LCV and higher cetane number. It was also noticed that it provided quieter and

smoother combustion than methanol operation.

(2) Maximum Fumigation Limit

An increase in the level of fumigation seems to be beneficial to engine performance, but it will increase the tendency of the engine to pre-ignite or knock. This condition is termed the maximum limit and represents a situation where continued operation of the engine is likely to result in damage to components.

From figures 5.5 and 5.6, it can be seen that the engine required high fumigation levels when the load or speed was low. The maximum fumigation level can go up to 80% with the assistance of inlet air heating to aid the vaporisation of fumigated alcohol. Compared to methanol, the engine required lower overall fumigation levels when running on ethanol.

V. INLET AIR HEATING LEVEL

Since the alcohol fuels have a high latent heat of vaporisation, it restricts their uses in compression ignition engines unless heating of the intake air is used to help vaporisation of the fuels.

Figures 5.7 and 5.8 show the required air temperature before it enters the carburettor. During this testing, the air heating level was controlled by a VARIAC auto-transformer. The temperature level was chosen according to the instability of the engine. High heating levels are used as little as possible to avoid a reduction in volumetric efficiency and unnecessary energy consumption of the heater.

Generally the heated air temperature changes substantially, while the temperature at the inlet manifold shows less fluctuation, especially for ethanol, see figure 5.9 and 5.10. In some cases, the manifold temperature is even higher than the heated inlet air temperature, e.g. the results obtained at 2000 rpm. This means that the manifold temperature plays a more important role in combustion than the heated air temperature. It indicates the temperature of the air/alcohol mixture and thus reflects the vaporisation of fumigated alcohol, the higher this temperature, the better the vaporisation of the alcohol fuel. It has a strong connection with engine operating conditions. Since the cooling water temperature of the test engine is maintained at 70°C and in most cases, the required manifold temperature remains around 40°C, it could easily be reached by means of a heat exchanger, alternatively an exhaust gas heated heat exchanger could be employed in practice.

VI. AIR/FUEL RATIO

The fumigation method may have a tendency for pre-ignition. If the fumigated fuel mixture is rich enough, pre-ignition might occur. Ethanol and methanol have lower limits of flammability of 3.5% and 5.5% (percentage by volume of gas in air), that is air/fuel ratios (by volume) are 27.6 and 17.2.

Figures 5.11 and 5.12 display the air/fuel ratio at maximum fumigation levels and different engine speeds. Only one point slightly falls down the lower limit of

flammability (figure 5.12). At that point, related manifold temperature is 36°C (refer to figure 5.9), which is far below the boiling temperature, hence some fraction of methanol condensed in inlet manifold. The fumigated mixture entered into the cylinder should be weaker than the calculated value, therefore the mixture didn't cause pre-ignition.

Methanol fumigation levels could be as high as to 83% when the maximum level was reached at 1500 rpm. This is still beyond the theoretical lower ignition limit. Therefore the fumigated mixtures would not cause any pre-ignition.

VII. COMBUSTION CHARACTERISTICS

From the literature, cylinder peak pressures are reported to reach an unacceptable level when conventional fumigation is employed. Maximum combustion pressure and the rate of pressure rise are two important factors that affect engine stress hence determining an engine's life-span and maintenance cost.

To determine in cylinder combustion pressures a pressure signal from a piezo transducer was amplified through a CUSSENS piezo channel unit and the data was stored in a print file in an IBM PC/XT compatible computer. Later it was transferred to a LOTUS 1-2-3 worksheet and printed out using an X-Y plotter. Figures 5.13 to 5.16 plot the pressure trace every 10° of crank angle for diesel and alcohol fuels. Later a new signal disk was designed to give a signal every 5° of crank angle. Therefore more

accurate curves were recorded as displayed in figures 5.17 and 5.18.

The engine was run over a range of conditions and data was collected at extreme operation conditions, i.e. from minimum fumigation and variable loads to maximum fumigation and various loads. Fumigation levels were adjusted from minimum, to just avoid the onset of misfire, to a maximum level without knock, using earlier experimental data as a guide.

From table 5.1, alcohol fuels have lower peak pressures compared to diesel fuels in most cases when the engine is running satisfactory. This becomes more obvious at higher speed. Fumigation doesn't have a sensitive effect on peak pressure for methanol, but peak pressure was recorded to be as high as 98 bar for ethanol at maximum load and low speed. Higher ethanol fumigation levels will deteriorate the running condition under the maximum load.

Ignition delays become more obvious at lower speeds. Figure 5.16 to 5.18 show ignition delays on the pressure traces at 1000 rpm when the engine operates on methanol and ethanol. This delay period contributes to the high pressure rate and higher peak pressure.

It is interesting to note there are two pressure peaks on the pressure traces. This is more likely to happen at reduced load and higher speed. This condition is not favourable for good combustion, even though fumigation does help auto-ignition of the injected fuel. After the first pressure peak, combustion pressure drops quickly then the remainder of injected fuel burns after some delay of

ignition and forms the second pressure peak. It is apparent that there is still considerable heat release once the fumigated mixture has ignited. This results in significantly higher cylinder pressure later in the expansion stroke compared to diesel only operation.

VIII. IGNITION DELAY

Ignition delay period is a very important parameter regarding CI engine operation. Fuel injection must be timed and metered exactly to control this parameter. It represents the time lag between the start of diesel engine injection and the first significant increase in cylinder pressure due to combustion. The ignition delay period includes the physical and chemical processes. The mechanism of fumigation is significantly different from the ordinary diesel operation. The alcohol fuel characteristics and the fumigated intake air will have effects on this ignition delay period.

For diesel only operation, ignition delay is not apparent at various running condition, see figure 5.13. This become more clearly visible in figures 5.15 to 5.18. The pressure signal was recorded every 5° crank angle in figures 5.17 and 5.18 after modification of the timing signal disk. This allows for more accurate description of combustion.

Figure 5.17 shows that the crank angle at ignition is about 5° BTDC with high fumigation level at 1000 rpm, while this is 0° TDC with minimum fumigation. Figure 5.18 reveals that the moment of ignition with minimum fumigation

for ethanol is later than that with maximum fumigation at 1000 rpm. From other traces at higher engine speed, e.g, the pressure trace at 2000 rpm, the point of ignition is not so clear, but the crank angle at peak pressure with maximum fumigation will occur earlier than that with minimum fumigation. Similar results were also obtained by Kwon et al (35), who showed that ignition delay was decreased by increasing the methanol quantity in the inlet air.

In figure 5.16, the ignition delay is greater and peak pressure at 1000 rpm is nearly 20 percent higher than that at 2000 rpm. This results in higher thermal efficiencies but at the same time, knocking is likely to occur due to the longer ignition delay period which is harmful to engine components.

Throughout these tests, the fuel injection timing was set at 30° BTDC and remained unchanged. From theoretical analysis, certain fixed ignition delay periods will cause ignition to occur later as the engine speed increases. In the real terms, the higher engine speed means that shorter time is available for heat losses from the engine cylinder, thus cylinder temperature will be higher and speed up the pre-combustion reactions of the fumigated charge. So the trend is for ignition delay to be reduced as engine speed and load increases.

From the results, thermal efficiency is found to be associated with ignition delay. As previously mentioned, low fumigation levels increase the ignition delay period and the peak combustion pressure will occur later in the

expansion stroke. Hence the mean effective pressure is reduced and this will contribute to a loss in thermal efficiency. This is clearly seen in figures 5.3 and 5.4. An area exists between the efficiency curves of the maximum and minimum fumigation levels. This simply means the higher the fumigation level, the higher the thermal efficiency, provided no knocking occurs.

IX. EFFECTS OF DIFFERENT SET-UPS ON ENGINE PERFORMANCE

From the baseline testing, engine set-ups are known to have an important impact on engine performance. Figures 5.3 and 5.4 show that fumigation has effects on both engine thermal efficiency and maximum power output. Injection timing also plays a role on engine operation.

From the literature, certain engine set-ups were reported to enable changes in engine operation and performance. Control parameters include:

- (i) Fumigation level of the alcohol/air mixture
- (ii) Injection timing of the alcohol fuel
- (iii) Compression ratio
- (iv) Preheating of the inlet air
- (v) Recirculation of exhaust gases

Amongst the above options, recirculation of exhaust gas has not been tested in this study because of the requirement for additional hardware and the fact that such recirculation increases particulate matter and carbon monoxide emissions. Since the diesel engine combustion process is very clean on alcohol fuels the implementation of such a system would be detrimental to the requirement

for a "clean" engine. Other of engine set-ups have been implemented on the Ricardo E6 engine with the following results.

(1) Fumigation Level

Engine loading was maintained unchanged, the alcohol injected into the cylinder was reduced by the auto-speed control unit while the fumigation level was increased. The engine performance versus fumigation is shown in figure 5.19. As the fumigation levels increase, there is a near linear relationship between the improvement in thermal efficiency and rate of fumigation. This improvement in efficiency is due to an improvement in indicated mean effective pressure as derived from pressure traces of the combustion process. It is believed that higher fumigated mixtures of alcohol/air will enhance the pre-combustion reactions, enabling the combustion process to complete more thoroughly, thus achieving higher efficiencies. Low fumigation levels mean, however, that more alcohol fuel is injected through the injector near the end of the compression stroke. This absorbs the compression energy of the fumigated mixture and takes a longer time to evaporate, followed by a longer ignition delay period. This less complete combustion leads to the lower efficiencies.

When fumigation level is higher than 35% of the total fuel flow, knocking occurs. The fumigated air/fuel ratio (by mass) is 20.3, this is still beyond the lower ignition limit. This contributes to a lower temperature at the end of the compression stroke and the subsequent longer ignition delay causes noisy combustion and knocking. Brake

power will drop slightly due to insufficient fuel supply through the injector when fumigation levels drops to zero.

In general, fumigation does have a positive role to play in improving engine performance. But careful control is required to avoid the extreme conditions of either misfire (power loss) at low level or knocking at high level.

(2) Injection Timing

At most conditions, the E6 engine cannot operate on 100% alcohol without fumigating alcohol into the inlet air. The main reason for this is that the alcohol fuel has a very high latent heat of vaporisation and high auto-ignition temperatures. This leads to an excessively long ignition delay period so that combustion will not occur.

The fumigation method supplies a charged mixture, which is sufficiently lean that it will not itself pre-ignition, and its temperature during compression will nearly reach its auto-ignition temperature. When the remaining alcohol is injected into the cylinder ignition will occur after compression and an ignition delay. Changes in injection timing could have an effect on the ignition delay period.

The results displayed in Figure 5.20 were obtained at settings of 1500 rpm with approximately 5% fumigation of methanol. The standard injection timing for diesel operation is 35° BTDC. Adjustment range is from 30° to 45° BTDC in the E6 engine. Engine thermal efficiency improves slightly with advancement of injection timing, whilst a constant level of brake power is maintained. Knocking

became very serious at 40° BTDC, The maximum advancement without knocking was only 37.5° BTDC. Advancing the injection timing forms a richer injected mixture this improves the pre-combustion reaction and achieves a higher efficiency. But on the other hand, abrupt ignition of a richer mixture causes a higher peak pressure and a rapidly increased rate of pressure rise which puts more stress on engine components and has such a noisy combustion that it must be avoided.

Retarding the injection timing will improve the noise of operation significantly with only a little lower thermal efficiency (3%) and lower brake power (5%). This is acceptable in exchange for smoother and quieter operating conditions. From the pressure versus crank angle diagram, the peak pressure can be seen to be close to the TDC position. For the following engine test, injection timing was set at 30° BTDC, this would protect the engine from noisy combustion.

(3) Heating Level

In a diesel engine, fuel is injected into the cylinder and ignites spontaneously. It unfortunately encounters the problem of cold starting. At the low cranking speed of an engine, compression doesn't begin significantly after BDC, when the intake valve closes. The effective compression ratio and thus the compression temperature, is therefore greatly reduced. The mixing of injected fuel and air formation is additionally unsatisfactory.

In the above process, the situation will greatly

deteriorate when diesel is replaced by the alcohol fuels because of their high latent heat of vaporisation. Therefore additional heating of inlet air will be beneficial to the overall temperature rise and hence ignitability. The fumigation method sometimes forms a rich mixture of vaporized fuel and air. This pre-mixture burns abruptly when it is mixed with injected fuel and produces the noisy combustion. This noise can be reduced by shortening the ignition delay by ways such as heating of inlet air.

Figure 5.21 shows the variation of thermal efficiency and brake power with T_{in} and T_m . Engine loading and the carburettor setting which controls the fumigation content were maintained unchanged. Thermal efficiency improved 4.2% as intake air temperature increased, while brake power had a decrease of 9.5%. There was also a slight decrease in engine speed.

The 1.5 kW electric heater contributes a high proportion of energy to warm the inlet air, but ends up with providing only a slight increase in thermal efficiency. Auxiliary heat will be more significant in improving the performance at lower engine loads and speeds. Power and speed were both reduced slightly, but the engine did run more smoothly with higher inlet air temperatures.

As the heat required to raise the air-fuel mixture in the inlet manifold is generally no greater than 60°C, this heat may be obtained by means of utilising a heat exchanger in either the exhaust or cooling water system. This will avoid the unnecessary extra energy consumption of

heating intake air electrically.

(4) Throttling

For the E6 engine, the inlet throttle adjustment is on the top of the inlet manifold but under the jet carburettor which is used to mix alcohol with inlet air. The indicating gauge could be changed from 0 to 100% open. This section therefore deals with throttling the mixture of air/alcohol fuel.

From figure 5.22, it can be seen that the effect of throttle is not obvious until the throttle is reduced to 60%. Here the air flow rate decreases rapidly and exhaust gas temperature and inlet manifold temperature increase markedly. The engine ceases to operate below 25% marked openness at 1500 rpm with an air flow rates drop of 22%; Also engine speed decreases when marked openness is 20% at 1000 rpm and then stops. Actual thermal efficiency shows a slight fluctuation of approximately 5% before a critical change of engine performance is observed. The engine power output, however, remains little changed.

Another comparative test was carried out at 1000 rpm, see figure 5.23. Engine performances were very similar at 100% and 65% openness of throttle, the latter had just a slightly less (2%) air flow rate. It seems difficult to make comparisons of engine performance with different throttle settings of fuel/air mixture because it is not sensitive to the inlet air flow rate. Therefore special hardware should be developed to control the inlet air throttle instead of the air fuel mixture throttle.

(5) Compression Ratio

At first, the engine was run at nearly full load condition at a speed of 1500 rpm with low fumigation levels. The compression ratio was reduced by means of the adjustment mechanism until the engine finally stopped after misfiring. This was expected because a lower compression ratio means lower compression pressure and temperature at the end of the compression stroke. Thus the combustion conditions deteriorated. This obviously is not to be recommended for the auto-ignition of low cetane number fuels, especially alcohol fuels with their very low cetane ratings.

Maximum engine output and the thermal efficiency decreased as the compression ratio reduced, figure 5.24. Inlet air temperature was raised to 89°C at a compression ratio of 17, otherwise the engine would have stopped because of unfavourable combustion conditions. The higher temperature helps the slight improvement in thermal efficiency.

X. DISCUSSION OF TRIAL RESULTS

Baseline test results reveal that the fumigation method can be applied to a diesel engine operating on near 100% alcohol fuels. Fumigation levels and auxiliary heating levels are both important in maintaining a stable engine operation.

The conclusions reached from the trial tests are as follows:

A diesel engine can run on nearly 100% methanol

except for the addition of 2% distilled water and 2% castor oil to the alcohol.

Engine performance is comparable to that obtained by diesel fuel only. Which is remarkable when one considers that the diesel engine has been developed exclusively to use diesel fuel for nearly a century. This is a most significant result from the study. The operating range reflects the flexibility for on road use, which could be limited because of the requirement for careful control of the fumigation level and heating level. This therefore requires this trial test to be followed by a further study with emphasis on a control system which will enable the current steady state operating conditions to be extended to cover transient operation. Further details will be discussed within the next chapter.

The engine was set up and operated on ethanol in a test programme similar in manner to that for methanol. Similar overall results were obtained and similar conclusions can be drawn. Therefore the future control system should be compatible to ethanol operation as well.

Fumigation alcohol plays an important role in diesel engine operation on alcohol fuels. It promotes the auto-ignition of low cetane number injected alcohol fuel. It improves the engine thermal efficiency as its content in the inlet air increases. It functions as an ignition forming centre and improves the thermal efficiency at low speed and reduced load conditions. The maximum level of fumigation is limited by the occurrence of knocking.

Auxiliary Heating of inlet air eases the

vaporisation of fumigated alcohol fuel. This creates a hotter charge which is useful in improving the flame speed and fuel ignitability, thus improving the engine performance. The negative effect is a reduction in volumetric efficiency of the engine when high temperatures are used.

Advancing injection timing seems to serve little use in trying to improve engine efficiency, but it does produce noisy combustion that can deteriorate the engine operating condition. For longer life of the research engine, retarding injection timing to 30° BTDC was adopted to achieve smooth operating conditions.

Higher compression ratios means higher compression pressure and temperature which is beneficial to the auto-ignition of low cetane number alcohol fuels. Engine output will be reduced with reduction in compression ratio, this is in agreement with theoretical analysis.

Fuel Pressure load		Diesel	Meth- anol (nor- mal pump)	Methanol (new pump)				Ethanol (new pump)			
				min. fumi.		max. fumi.		min. fumi.		max. fumi.	
				bar	% *	bar	% *	bar	% *	bar	% *
min. load	1000rpm	65	64	60	51.6	56	84.5	65	31.5	62	32.8
	1500rpm	61	53.5	46	0	51	79.8	49.5	14.8	48.5	57.6
	2000rpm	63	56	46	0	46	12.2	53	18.9	52	40.4
max. load	1000rpm	71.5	62.5	60	11.1	60	15.3	76	31.5	98	38.6
	1500rpm	63.5	50	48.5	0	50.5	13.4	56	.45	82	33.8
	2000rpm	63	60	48.5	12.5	47.5	12.5	59	1.18	55	13.7

* fumigation level by percentage.

Table 5.1. Effects of fumigation and load on peak pressure

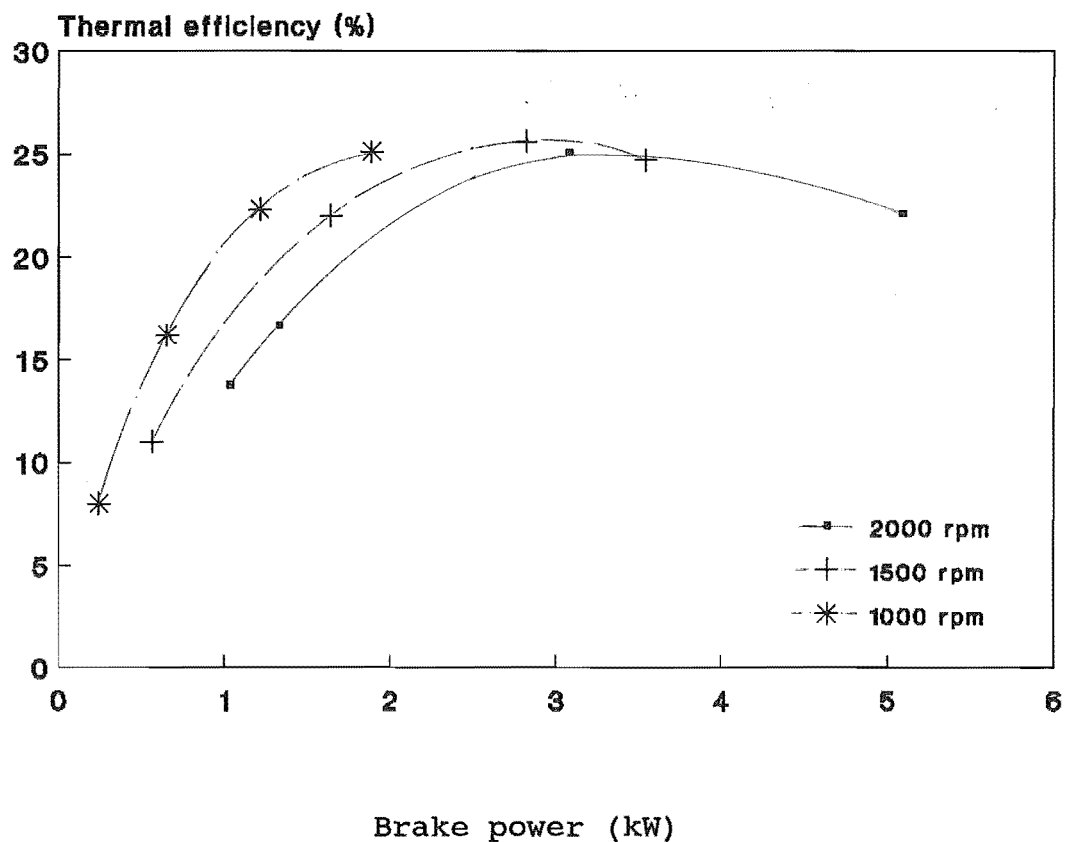


Figure 5.1 Engine performance on diesel

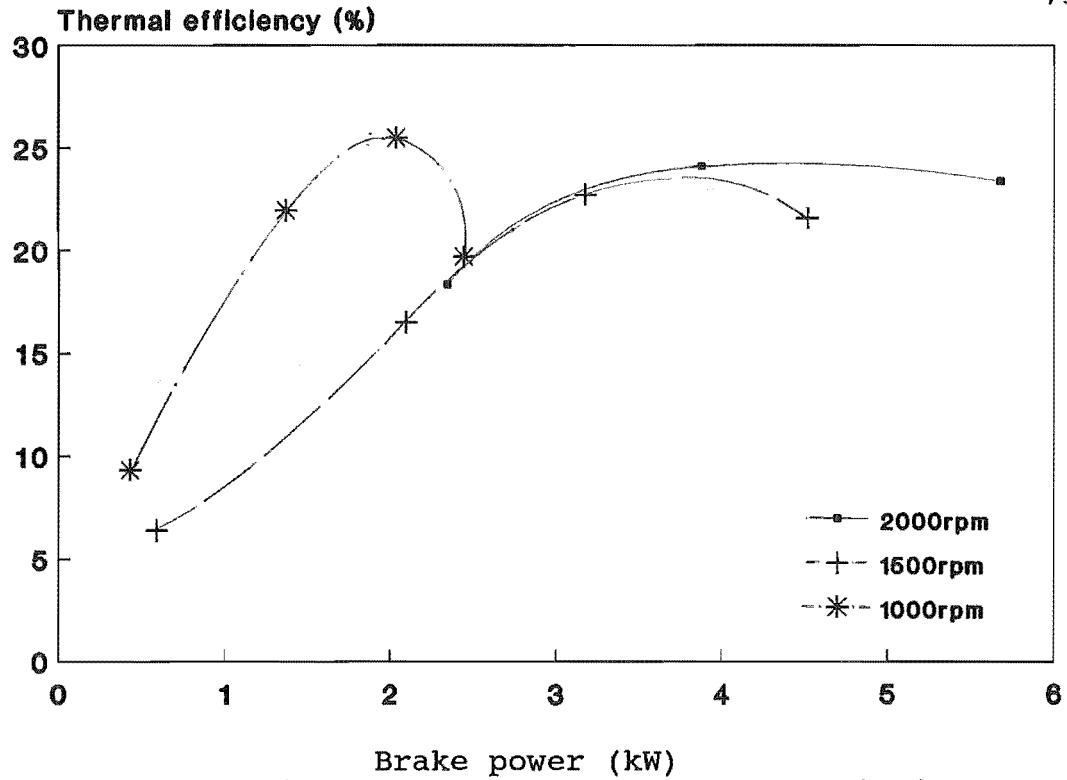


Figure 5.2(a) Engine performance on methanol(existing pump)

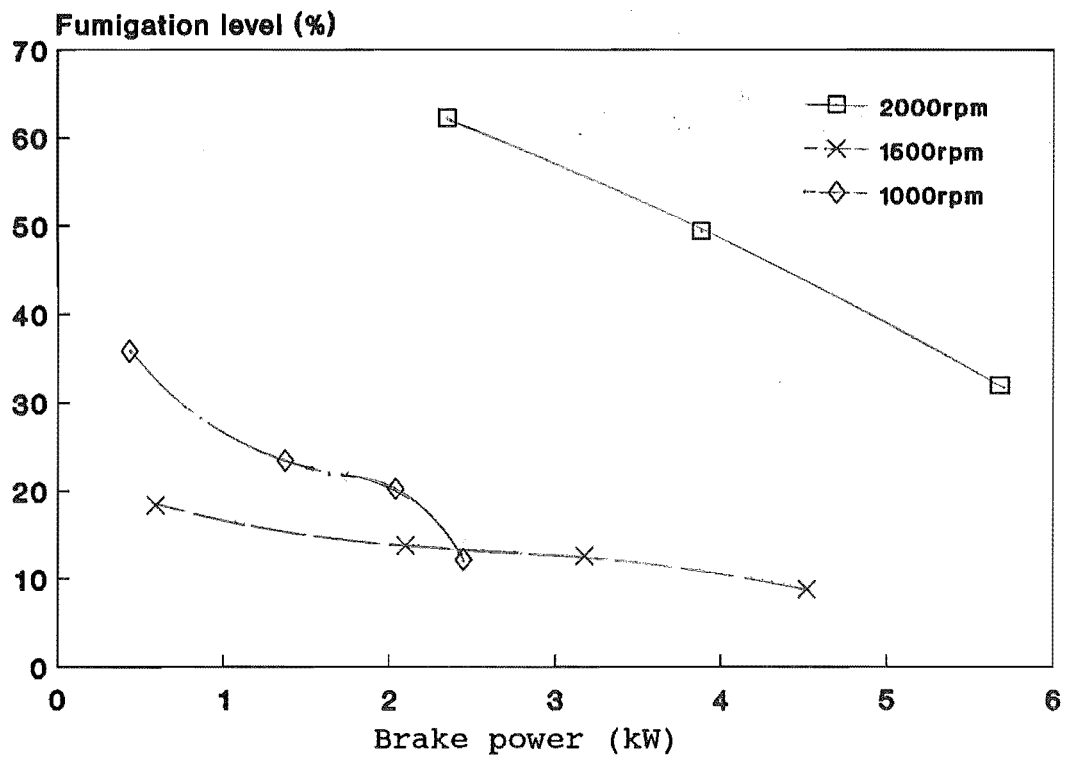


Figure 5.2(b) Fumigation level(methanol, existing pump)

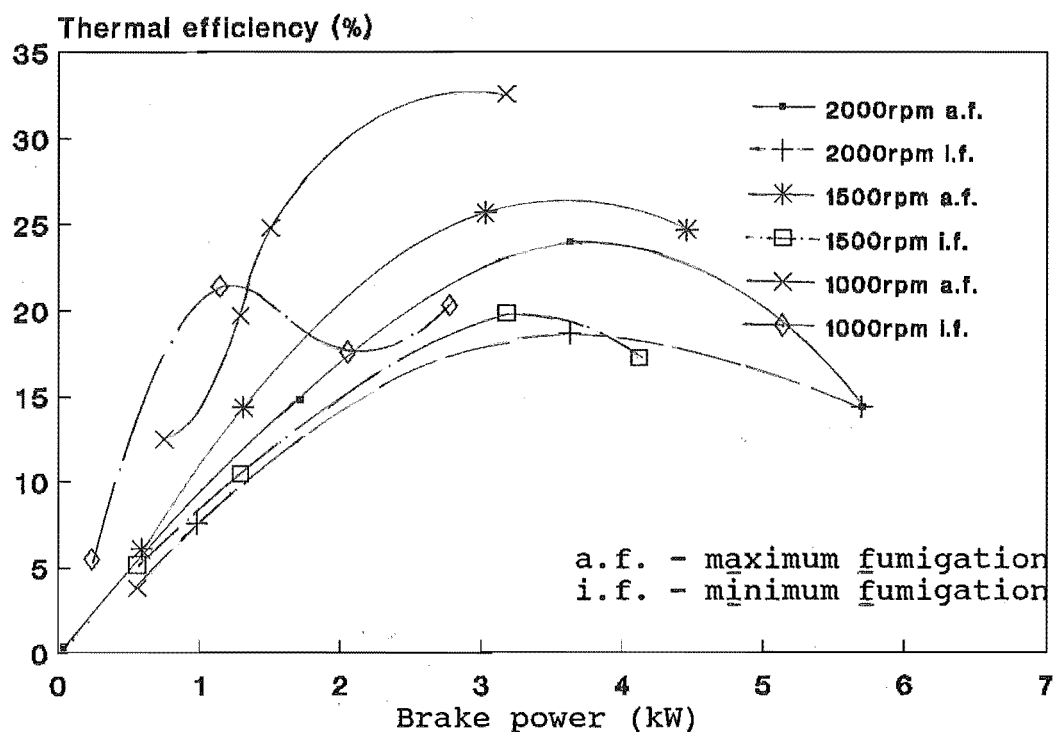


Figure 5.3 Engine performance on methanol

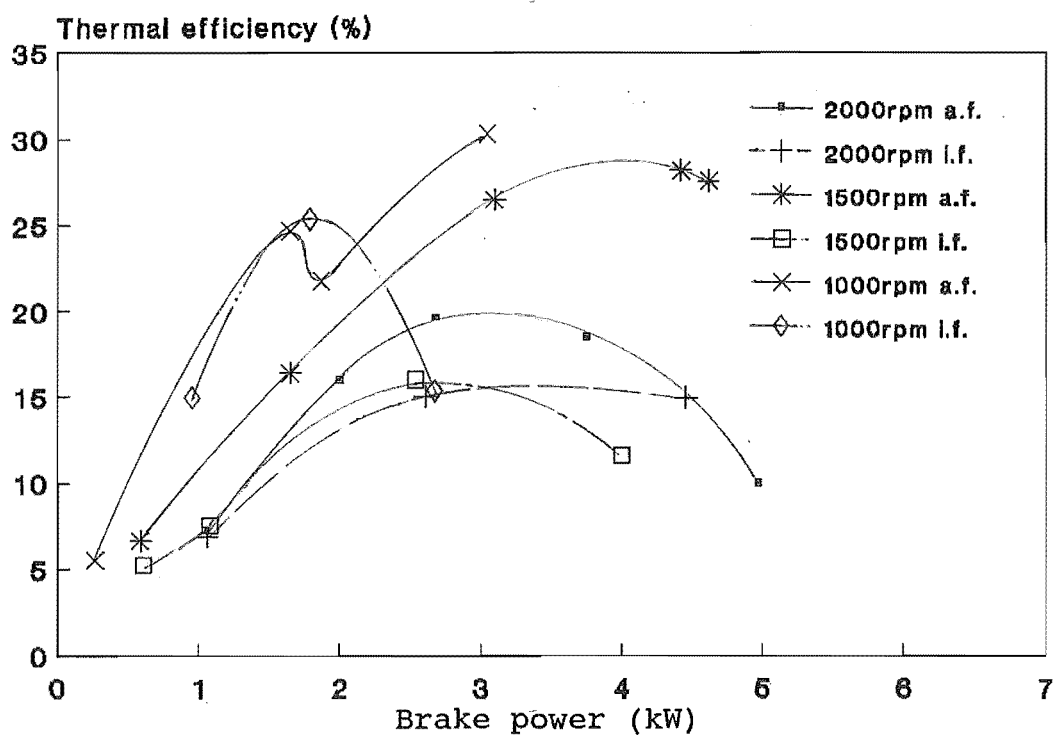


Figure 5.4 Engine performance on ethanol

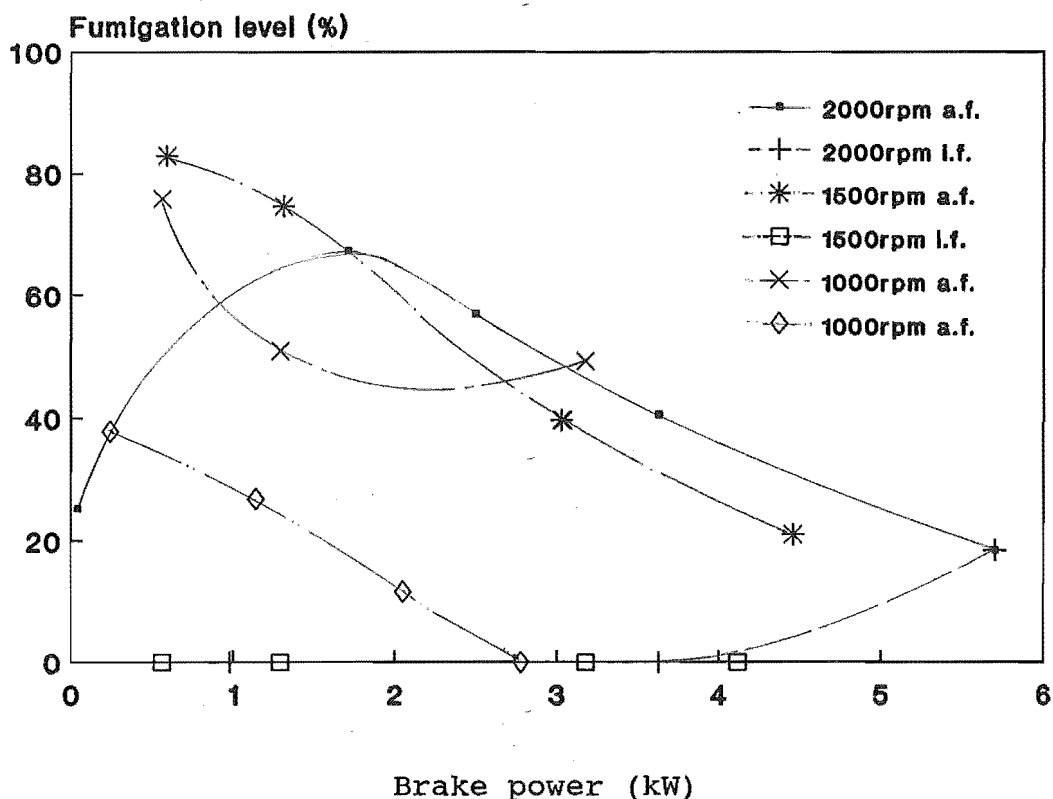


Figure 5.5 Fumigation level (methanol)

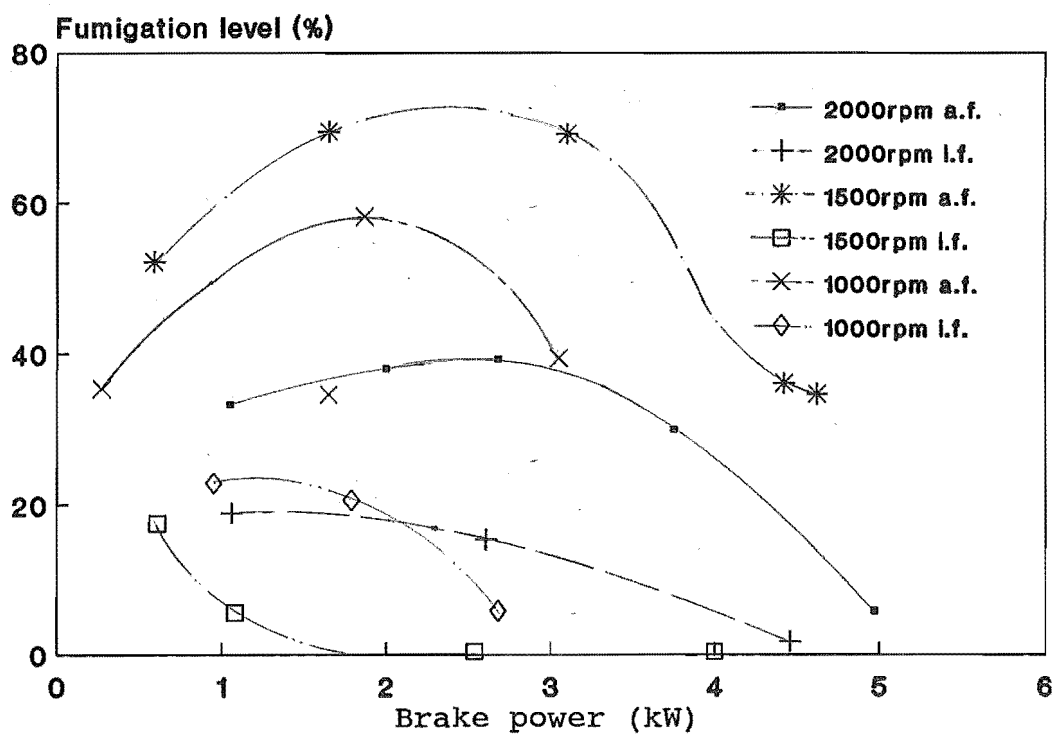
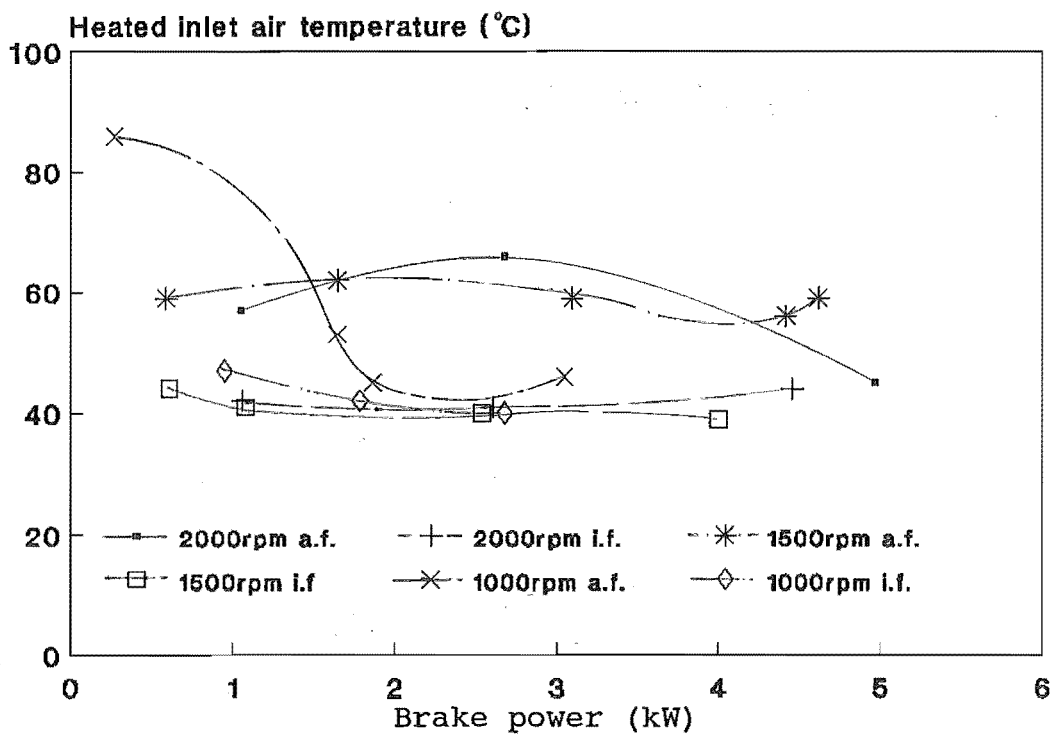
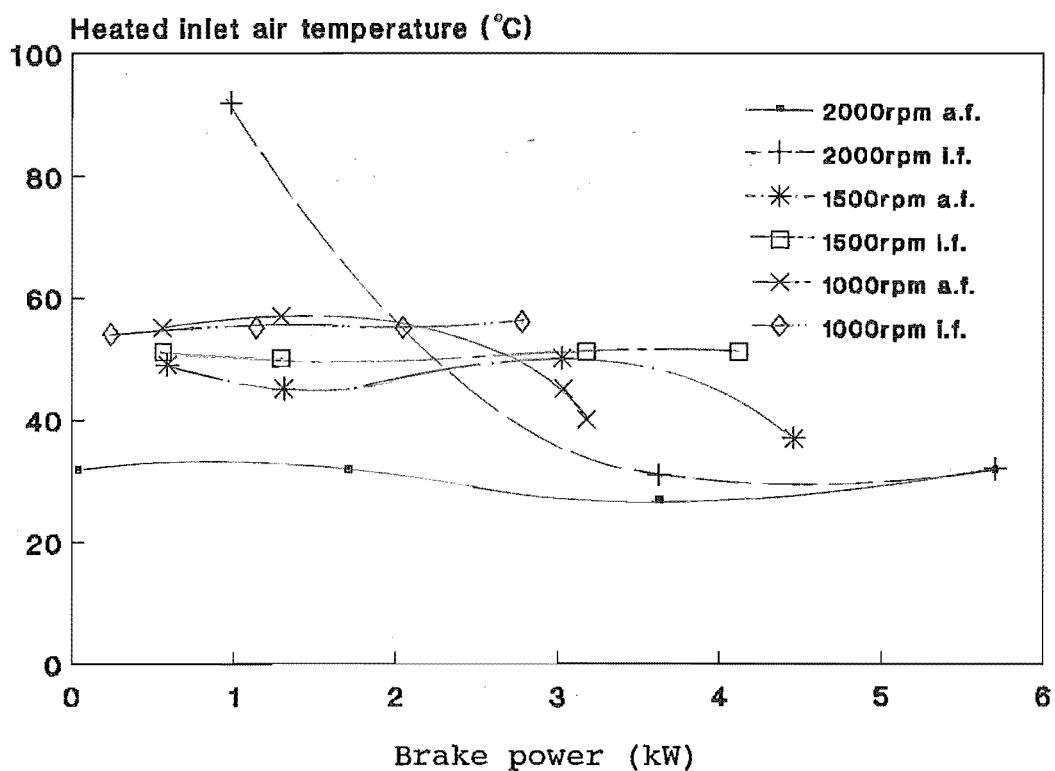


Figure 5.6 Fumigation level (methano)



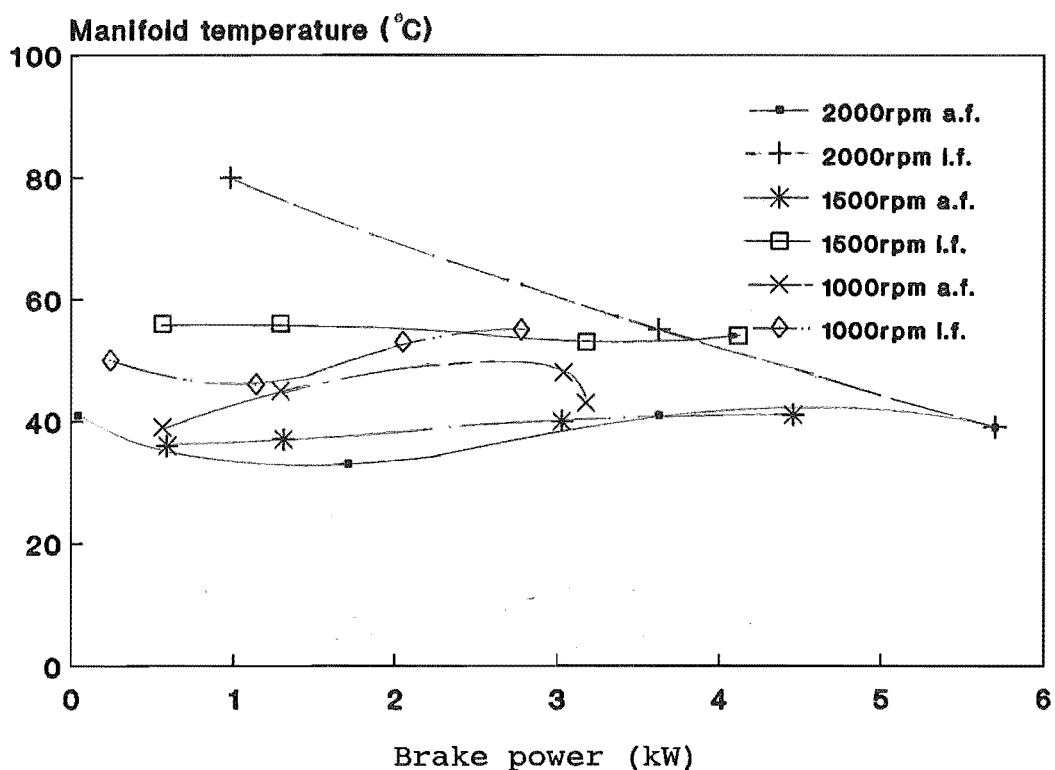


Figure 5.9 Manifold temperature (methanol)

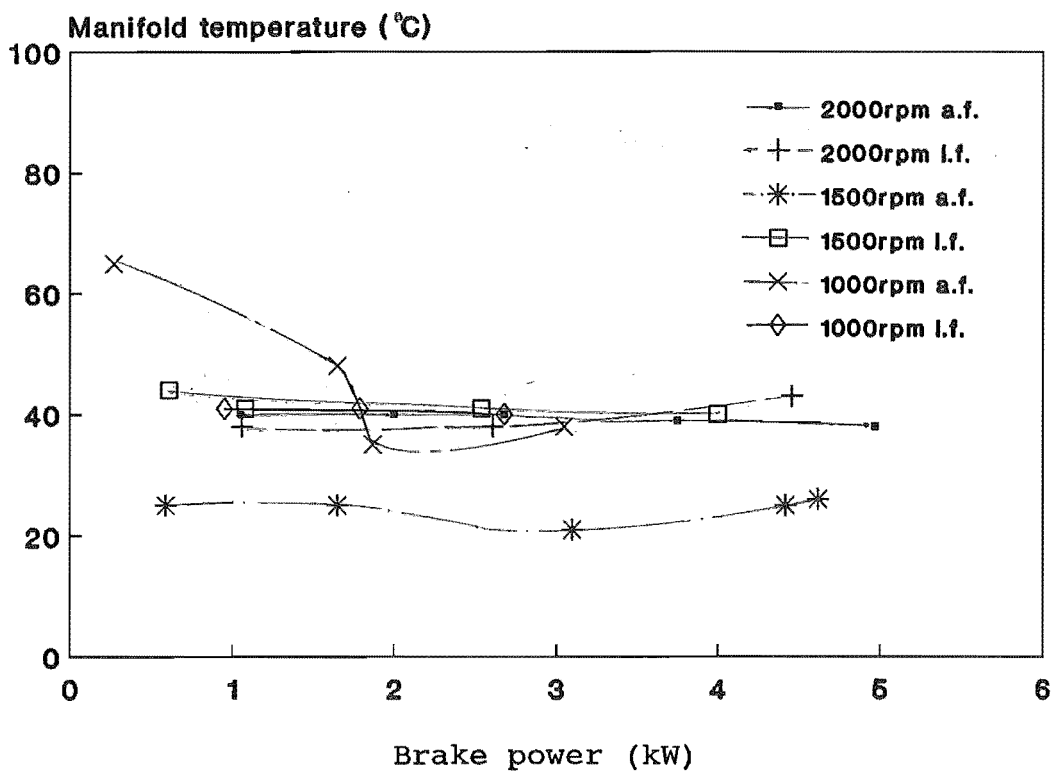


Figure 5.10 Manifold temperature (ethanol)

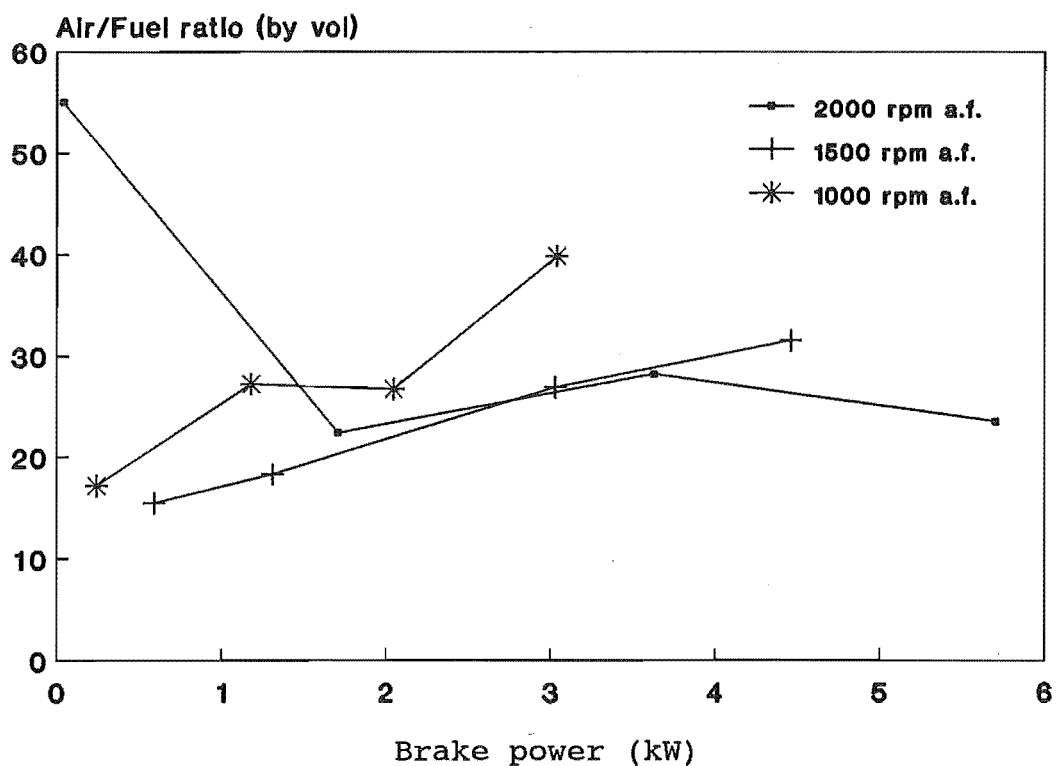


Figure 5.11 Air/fuel ratio of fumigated mixture (methanol)

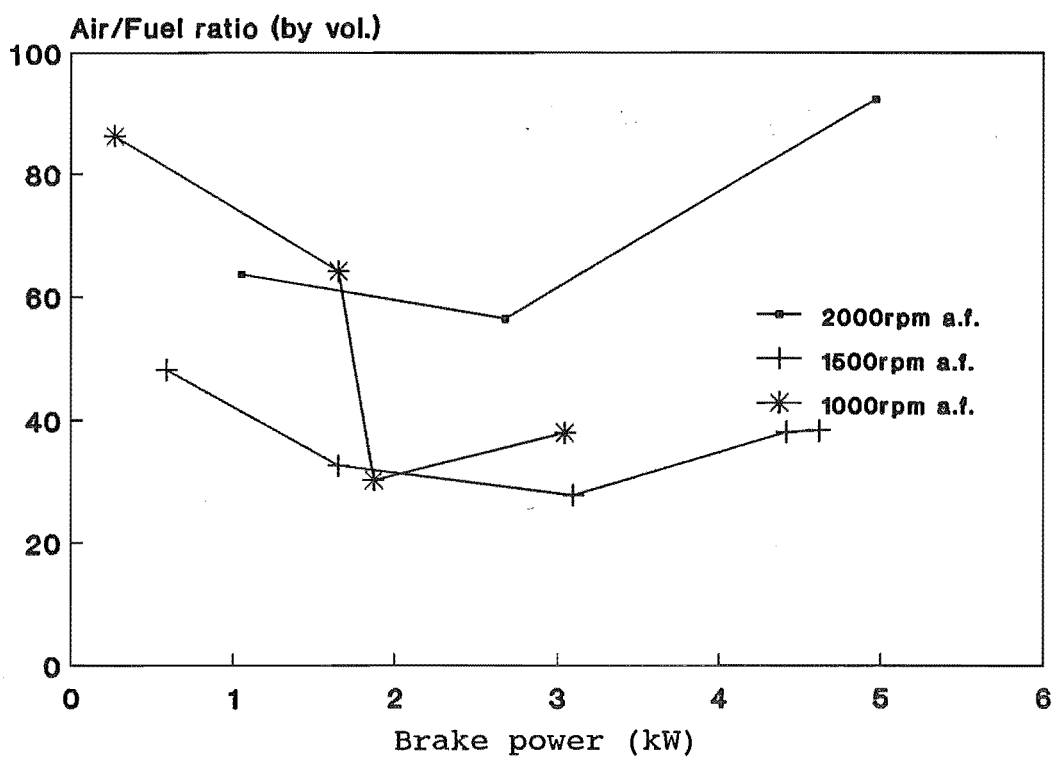


Figure 5.12 Air/fuel ratio of fumigated mixture (ethanol)

Fig. 5.13(a) Pressure vs Crank Angle
 - min. load, diesel -

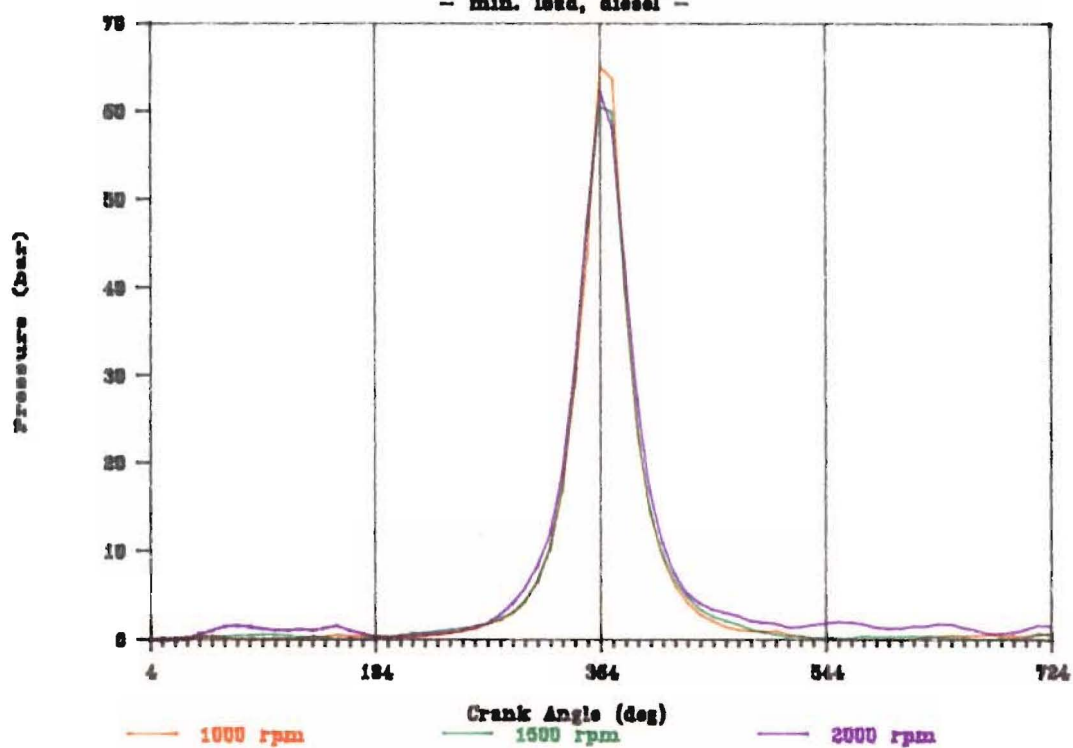


Fig. 5.13(b) Pressure vs Crank Angle
 - max. load, diesel -

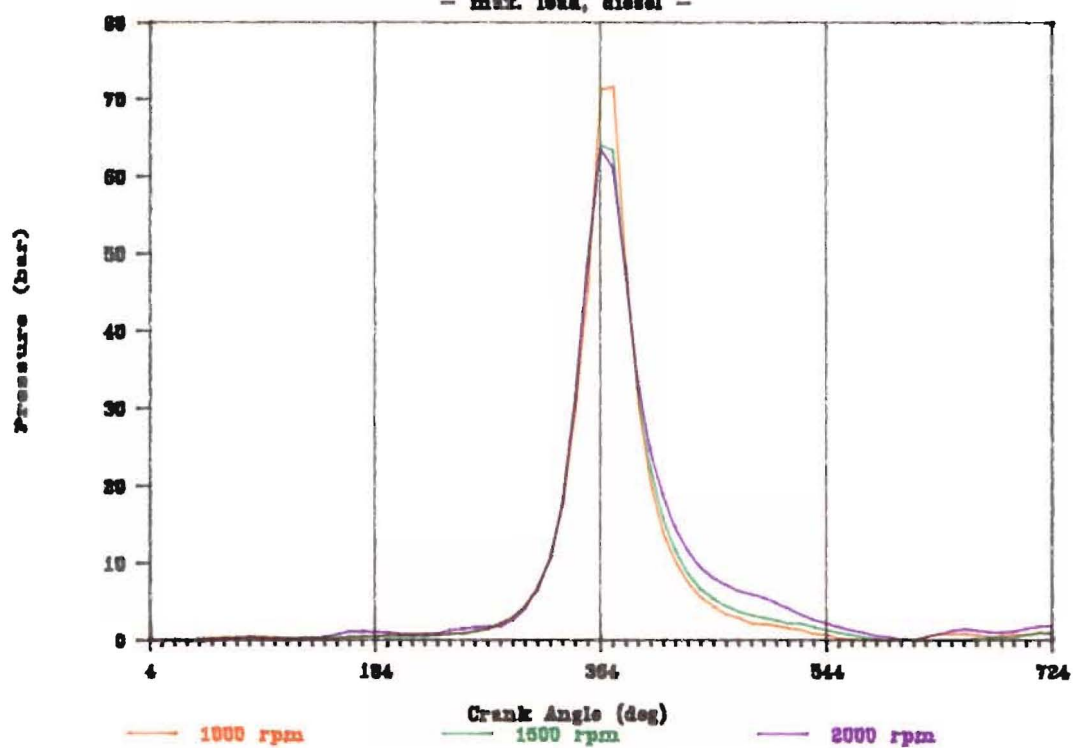


Fig. 5.14(a) Pressure vs Crank Angle

- min.load, methanol, normal pump -

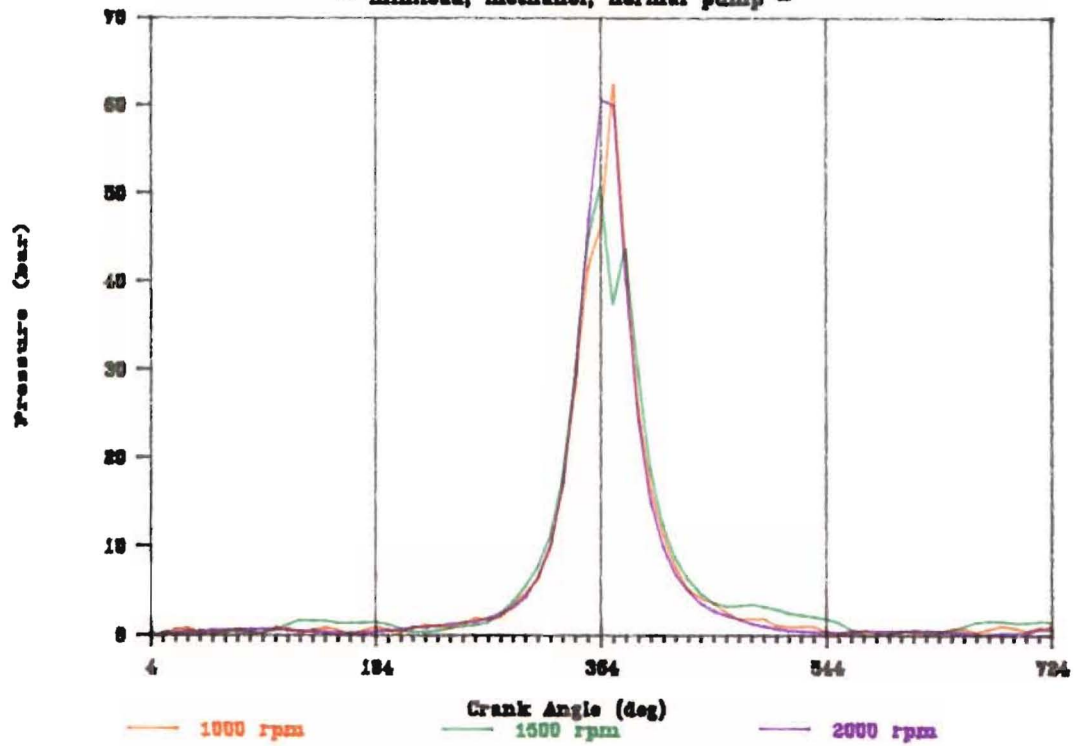


Fig. 5.14(b) Pressure vs Crank Angle

- max.load, methanol, normal pump -

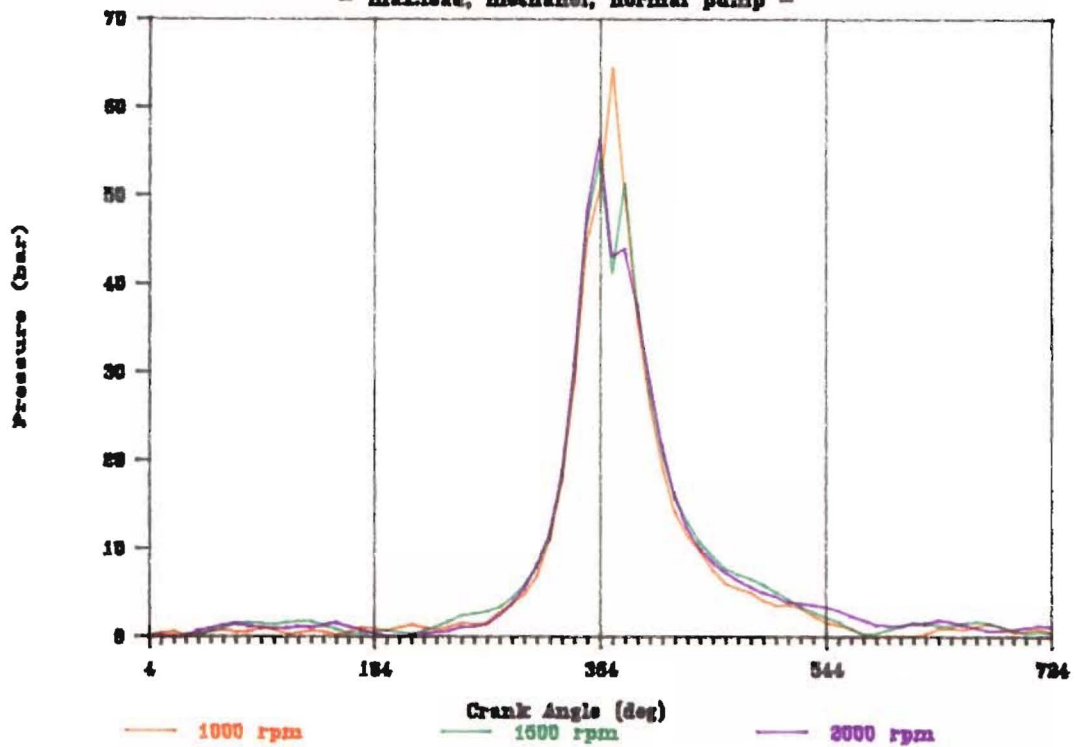


Fig. 5.15(a) Pressure vs Crank Angle

- min.load, min.fam. methanol -

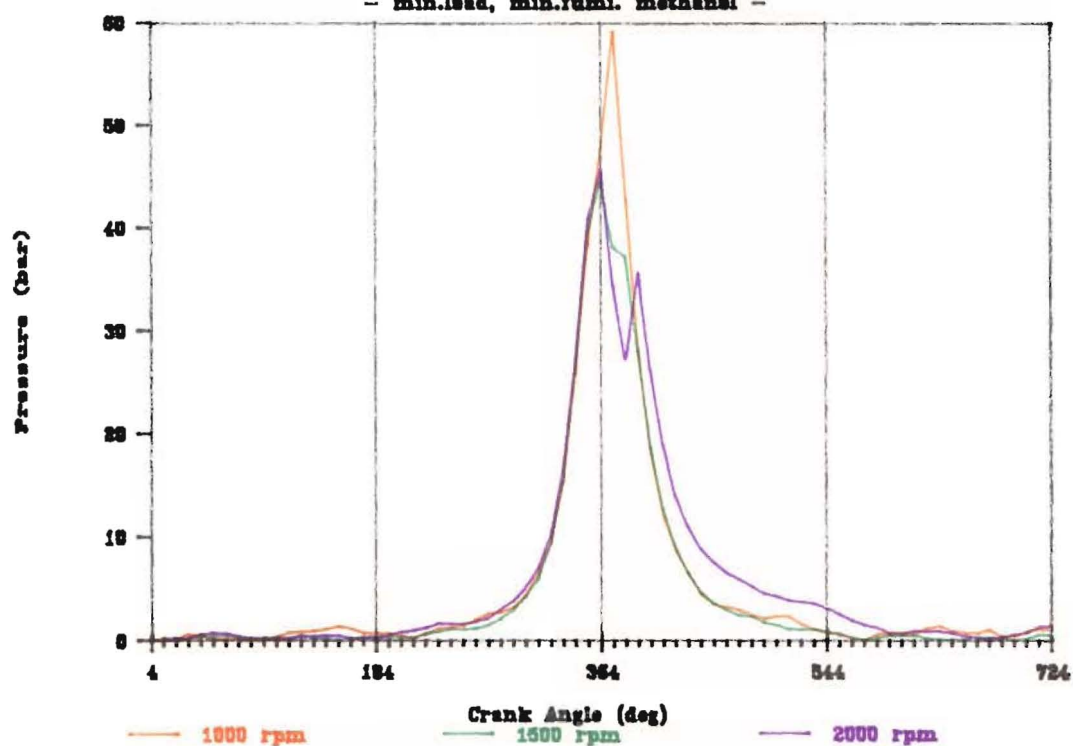


Fig. 5.15(b) Pressure vs Crank Angle

- min.load, max.fam. methanol -

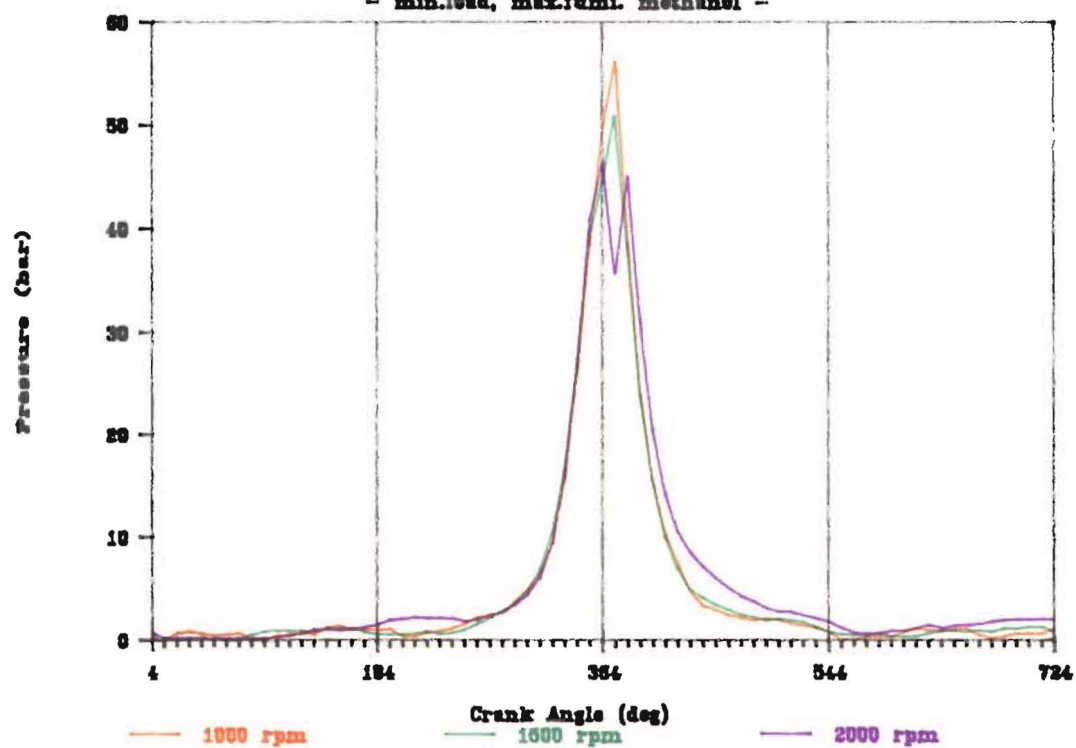


Fig. 5.16(a) Pressure vs Crank Angle

- max.load, min.fuml. methanol -

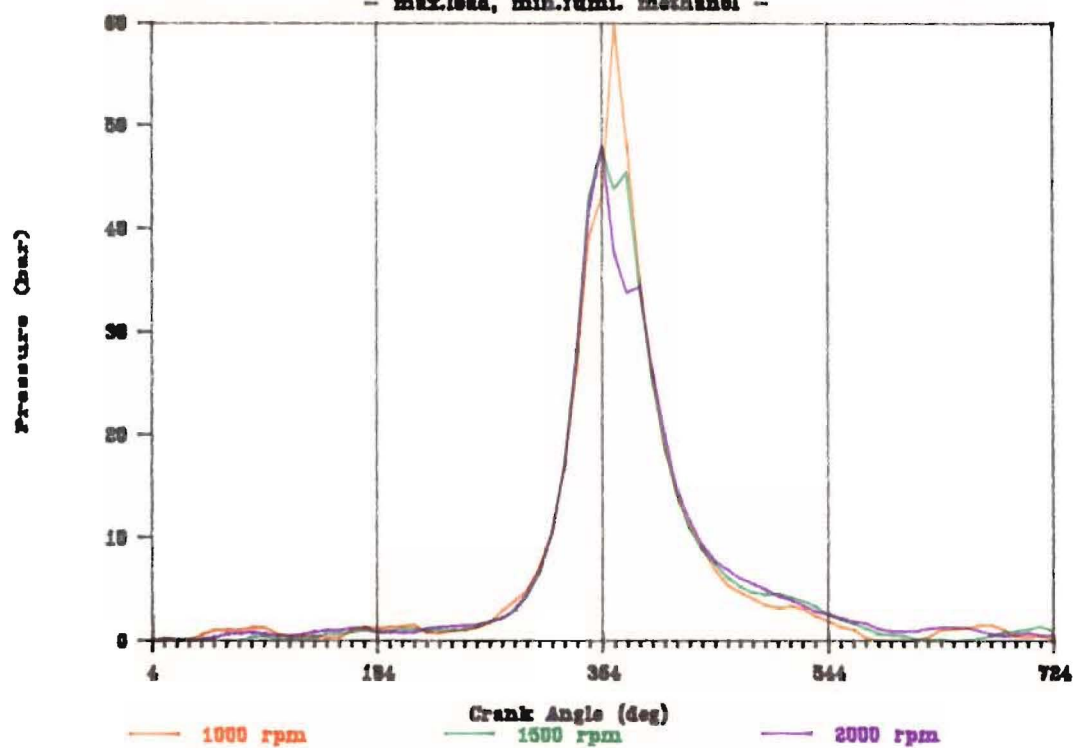


Fig. 5.16(b) Pressure vs Crank Angle

- max.load, max.fuml. methanol -

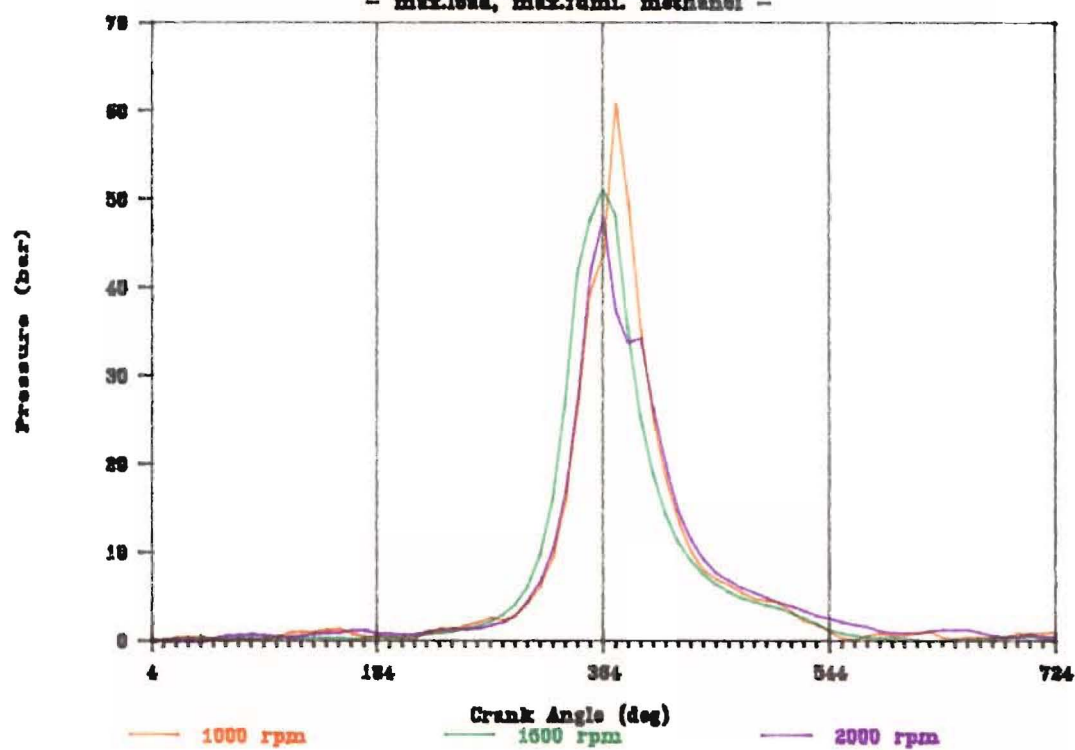


Fig. 5.17(a) Pressure vs Crank Angle

- min.lead, min.fuel, ethanol -

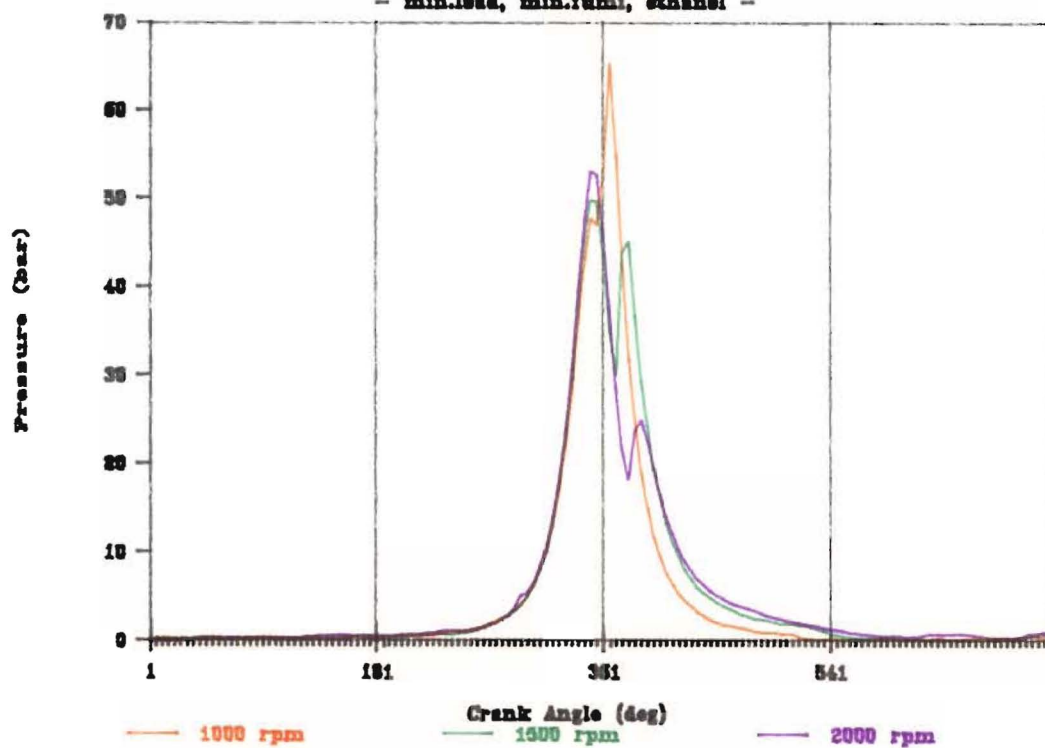


Fig. 5.17(b) Pressure vs Crank Angle

- min.lead, max.fuel, ethanol -

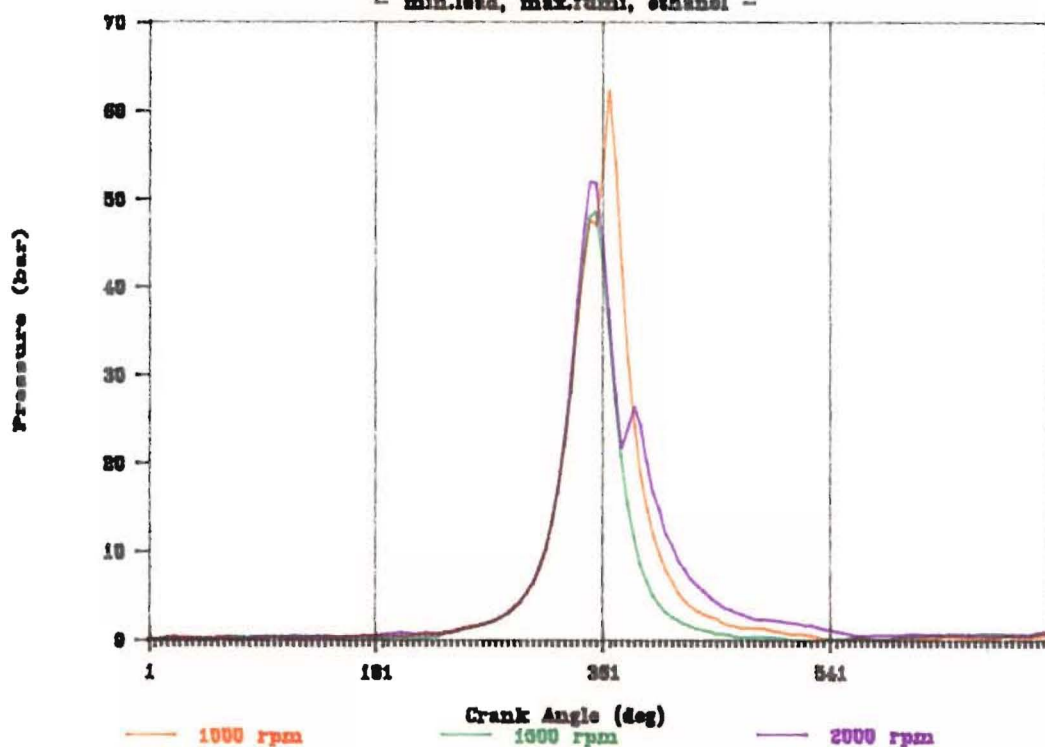


Fig. 5.18(a) Pressure vs Crank Angle

- max. load, min.fum. ethanol -

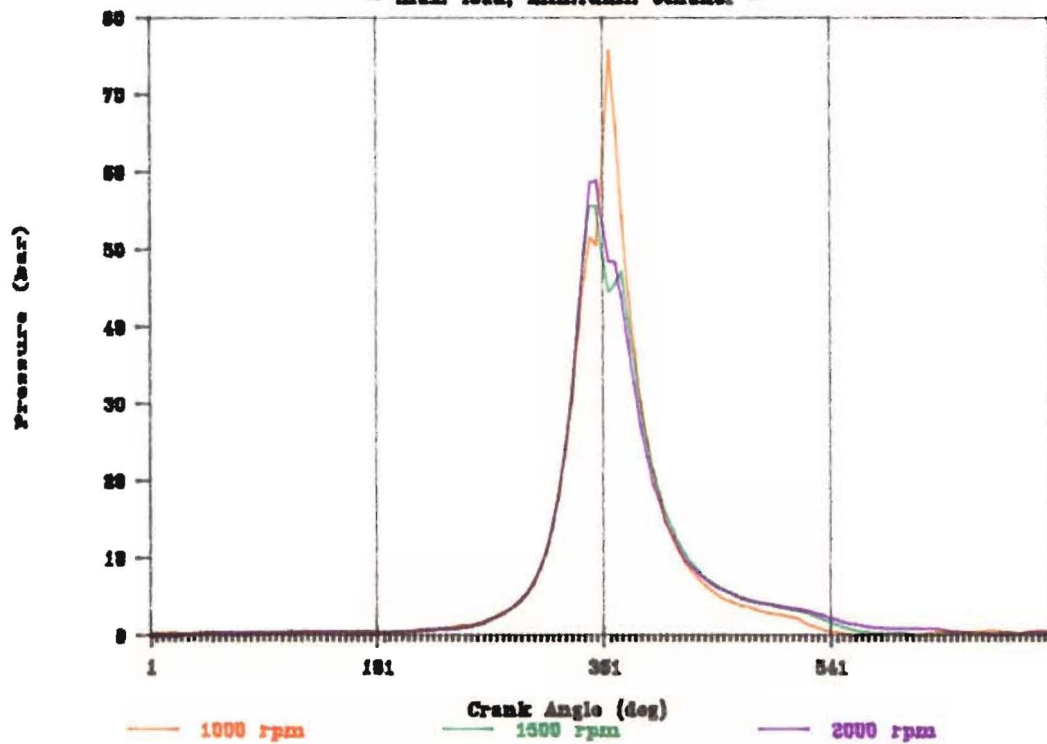
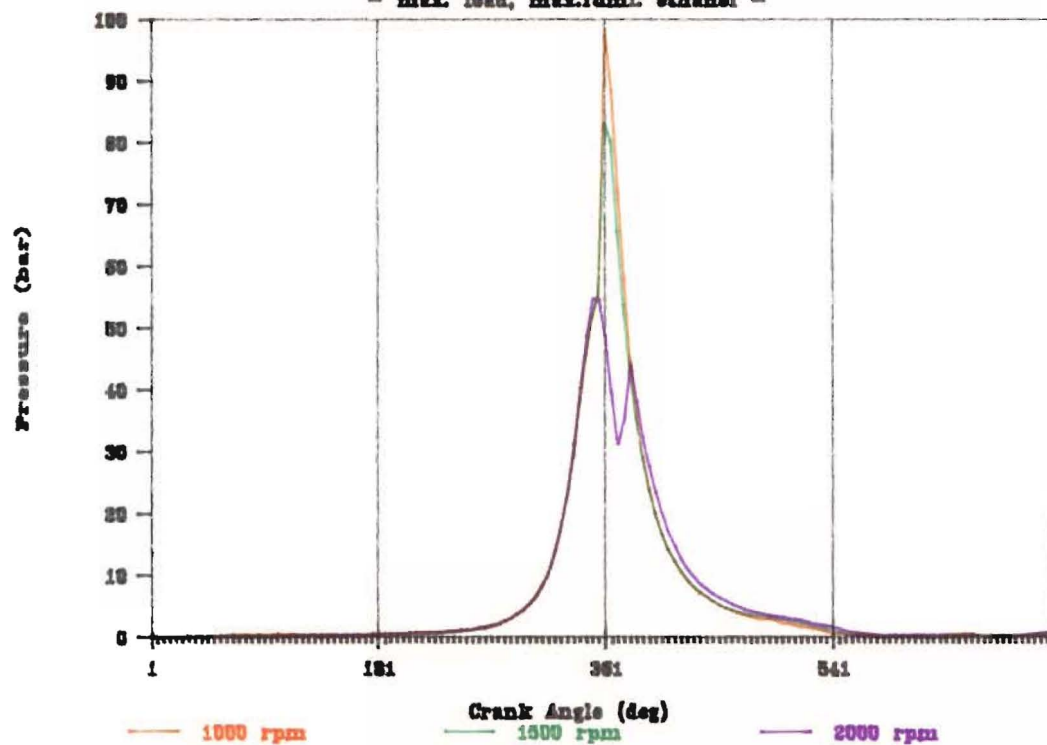


Fig. 5.18(b) Pressure vs Crank Angle

- max. load, max.fum. ethanol -



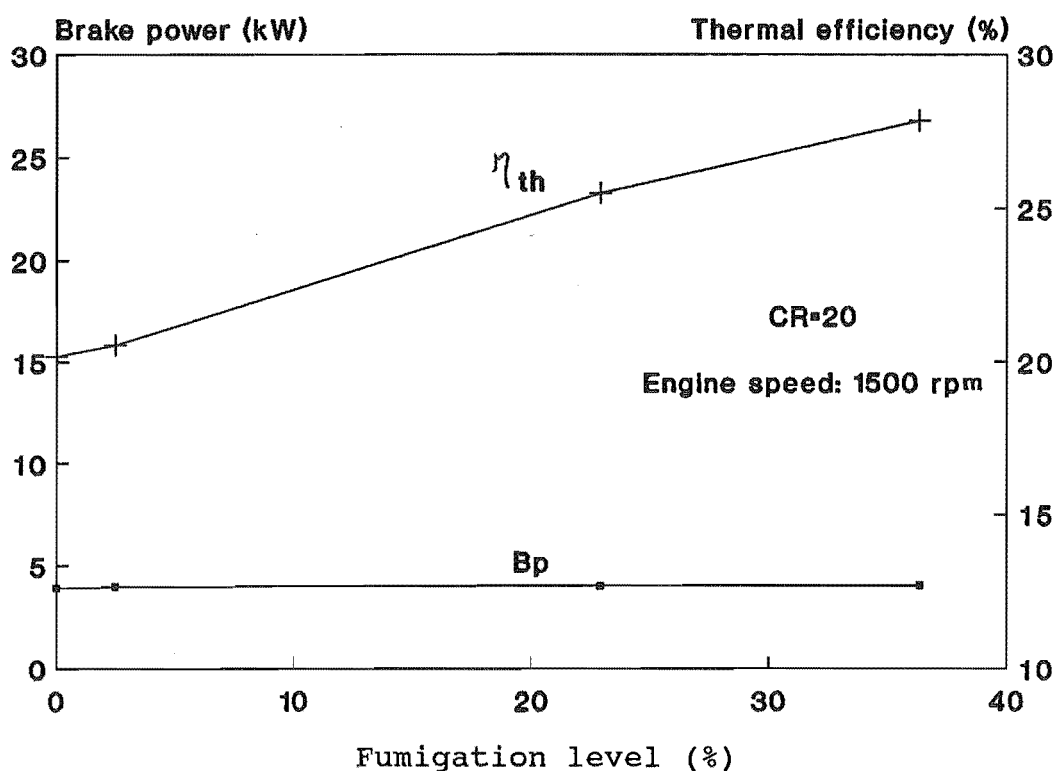


Figure 5.19 Performance vs fumigation level (methanol)

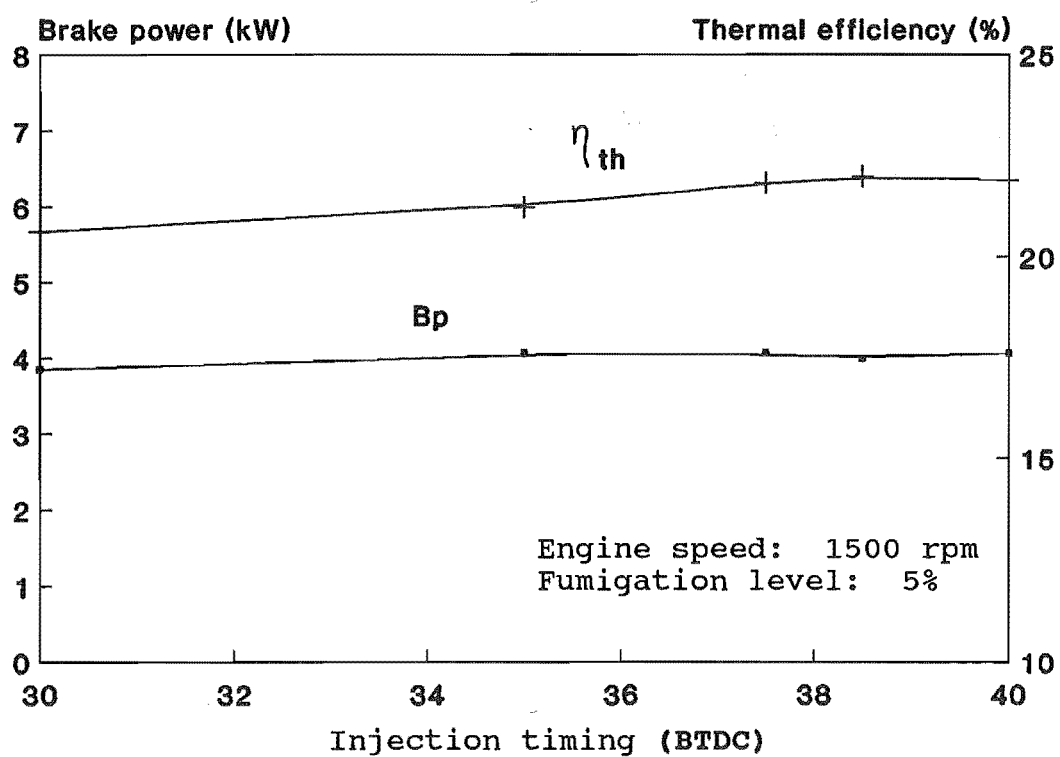


Figure 5.20 Performance vs injection timing (methanol)

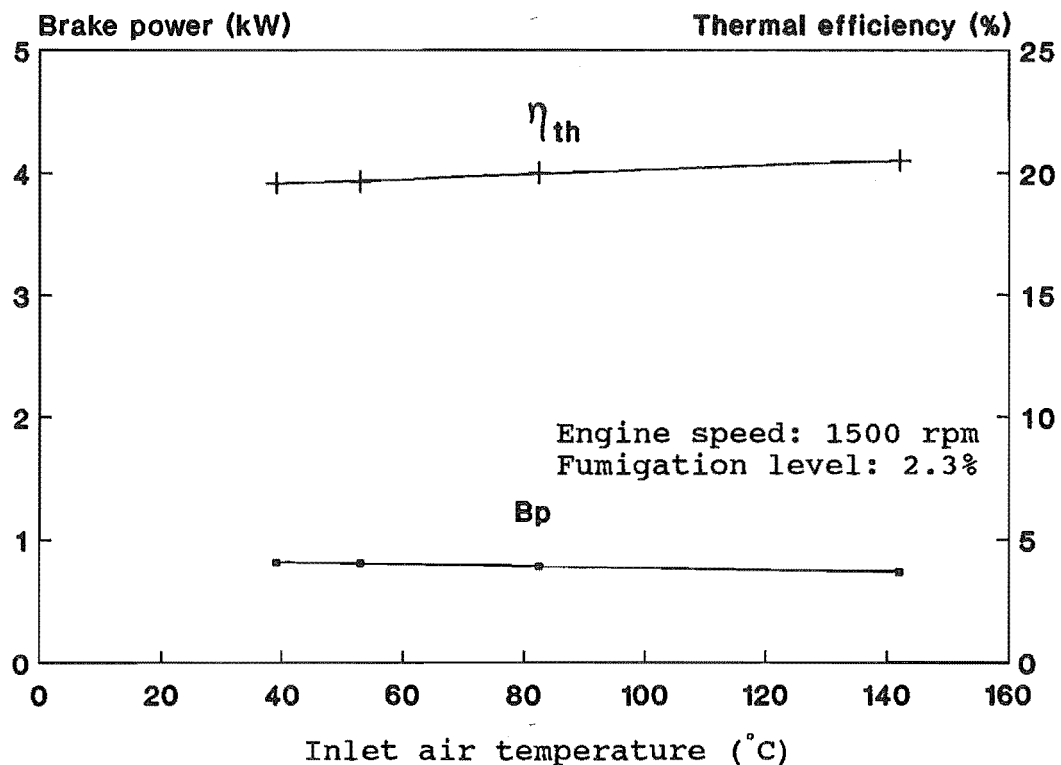


Figure 5.21(a) Performance vs inlet air temperature(methanol)

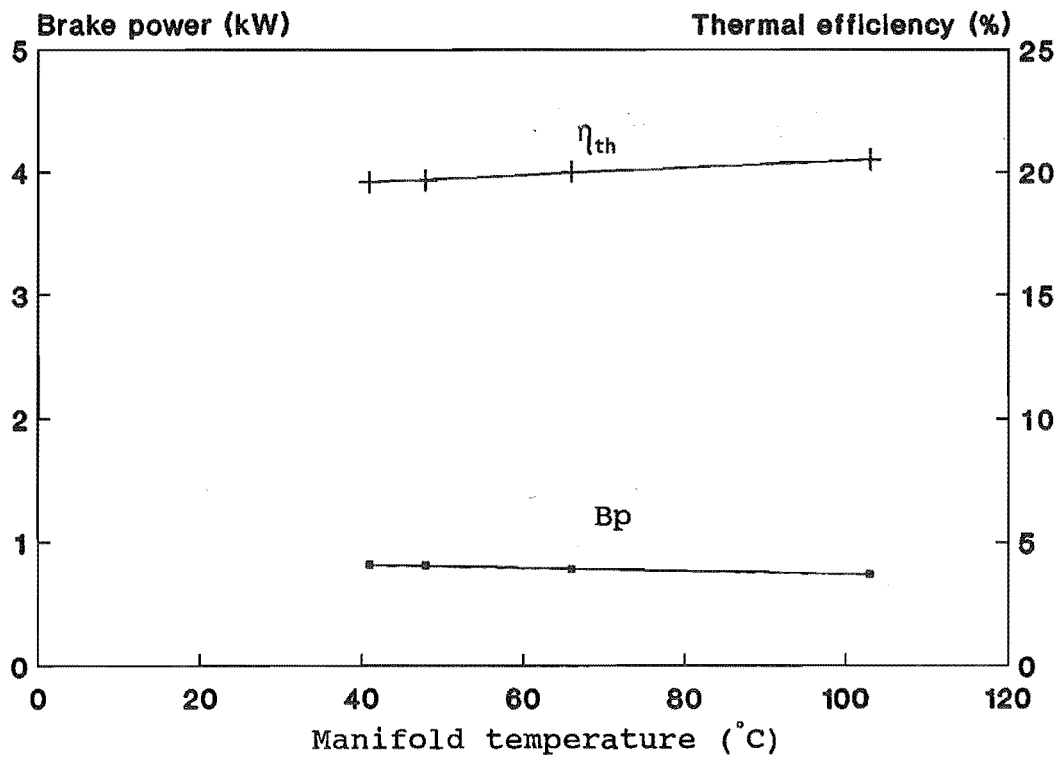


Figure 5.21(b) Performance vs manifold temperature

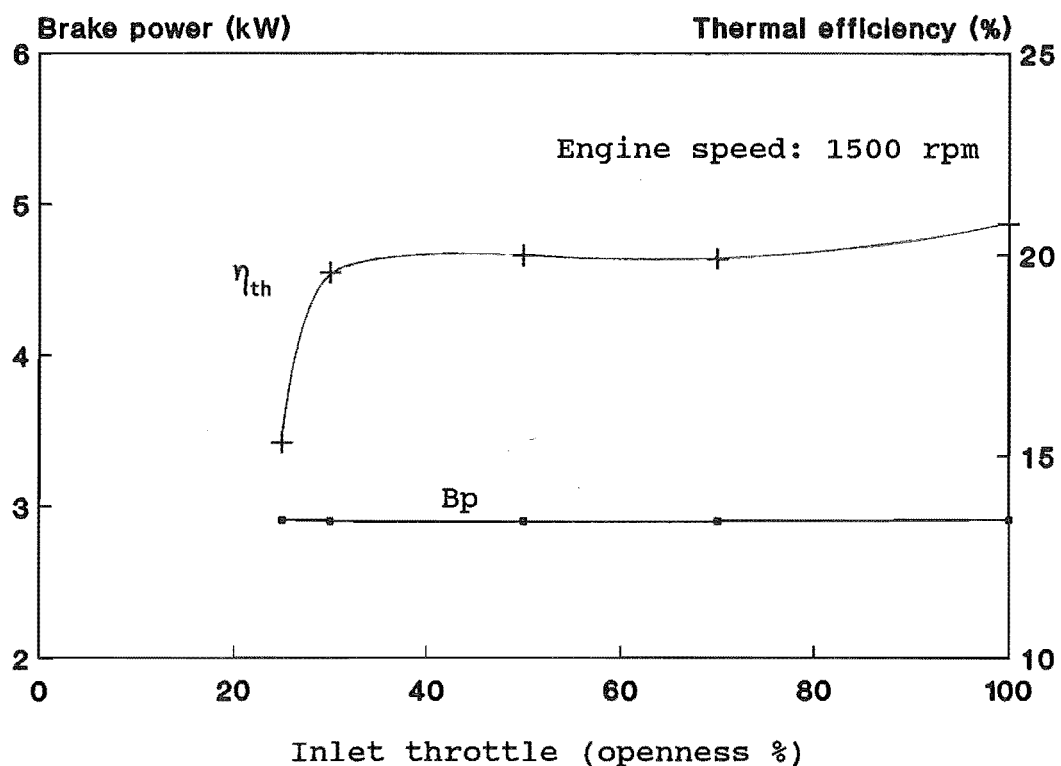


Figure 5.22(a) Performance vs inlet throttle (methanol)

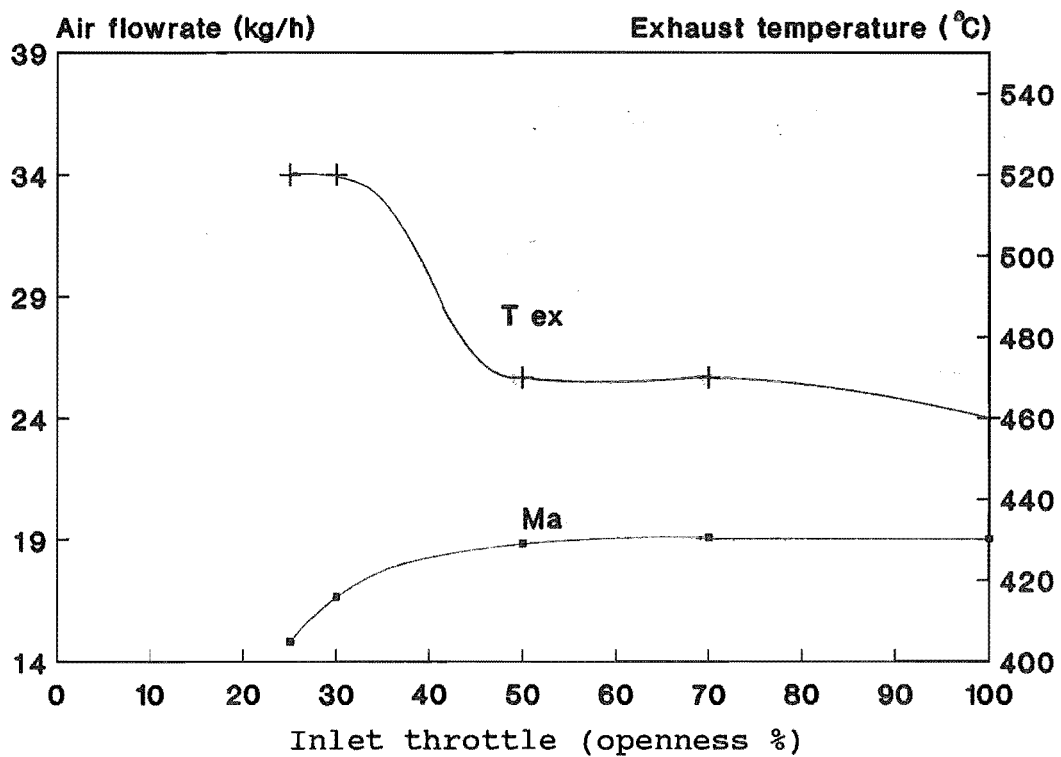


Figure 5.22(b) Effect of inlet throttle

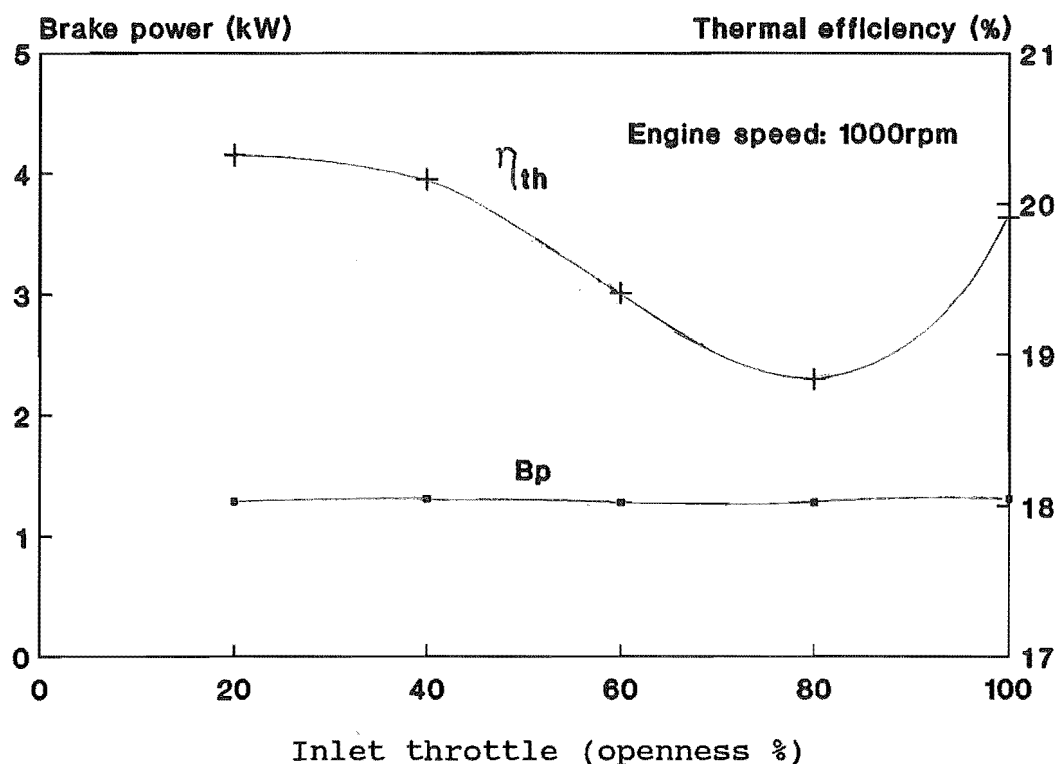


Figure 5.23(a) Performance vs inlet throttle (methanol)

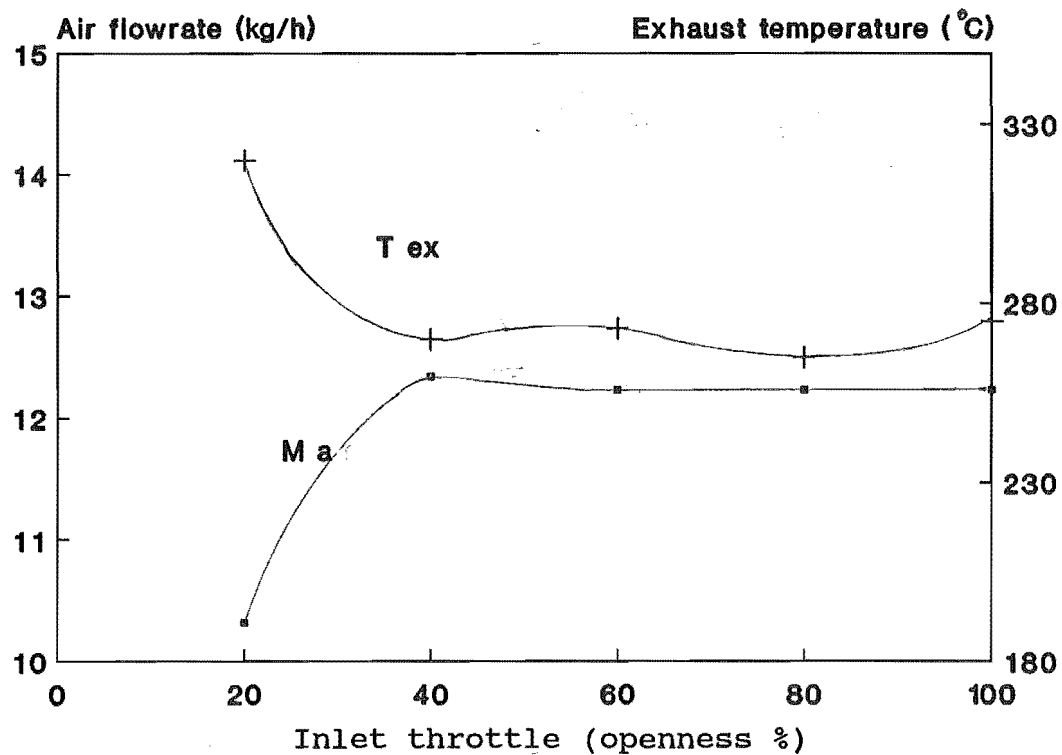


Figure 5.23(b) Effect of inlet throttle

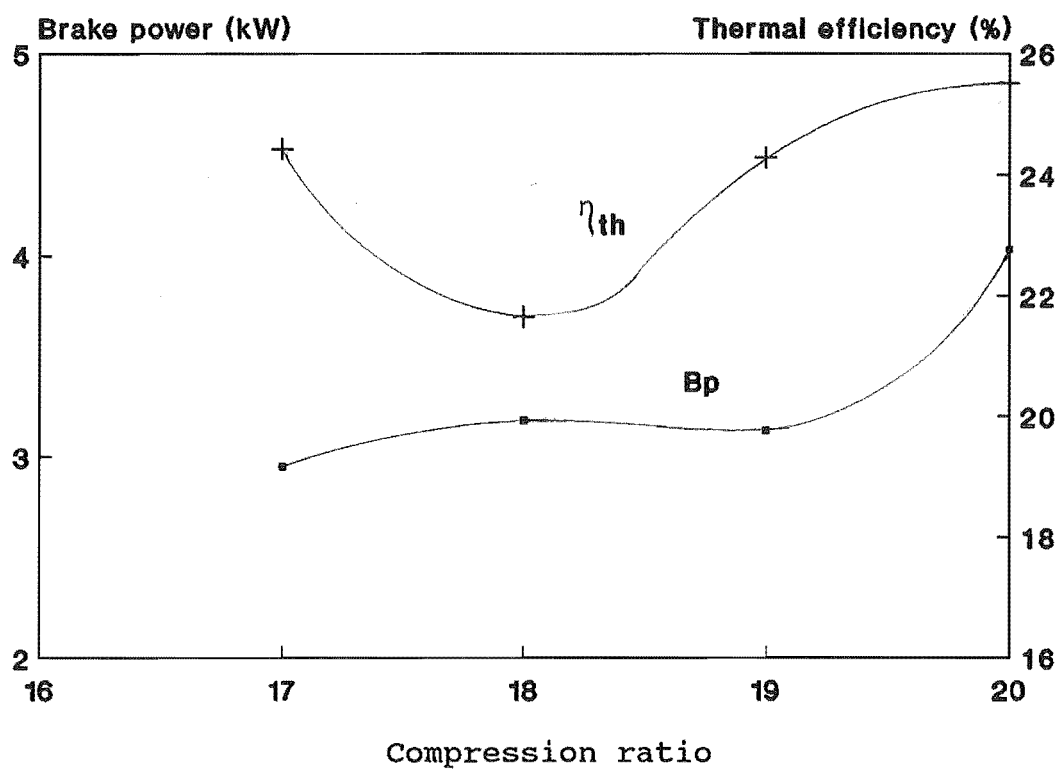
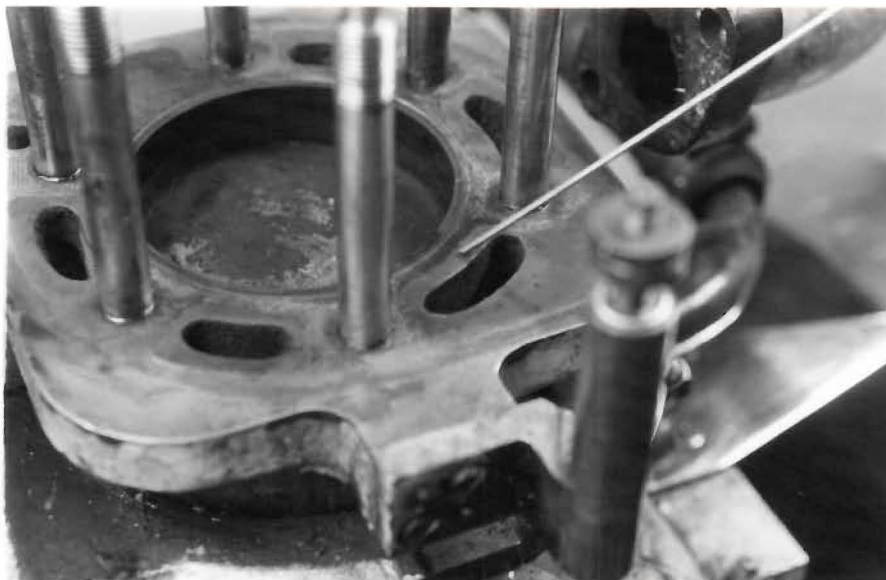


Figure 5.24 Performance vs compression ratio



(a)



(b)

Plate 5.1 Broken cylinder head gasket

CHAPTER VI

CONTROL SYSTEM

As a consequence of the baseline test results, an accurate control system is required to achieve optimal engine performance under steady state and transient conditions. Traditional experimental methods are tedious, time consuming and often lack the accuracy and repeatability that scientific research requires. A computer based data acquisition and control system is of very great practical interest to this study.

I. TEST SIGNAL SUMMARY

In chapter IV the test equipment was described. Following shows the hardware selected for measuring the engine parameters in chapter IV and the signals to be presented to the data acquisition system.

Test signals can be summarized as follow:

Analogue (A)

Digital (D)

Pulse (P)

Voltage (V)

Current (C)

Resistance (R)

Engine parameters for data acquisition include:

(a) Temperature

(b) Pressure

(c) Speed

- (d) Trigger and pulse counting
- (e) Engine torque
- (f) Fuel flow

The relationship between data acquisition and signals are shown in figure 6.1.

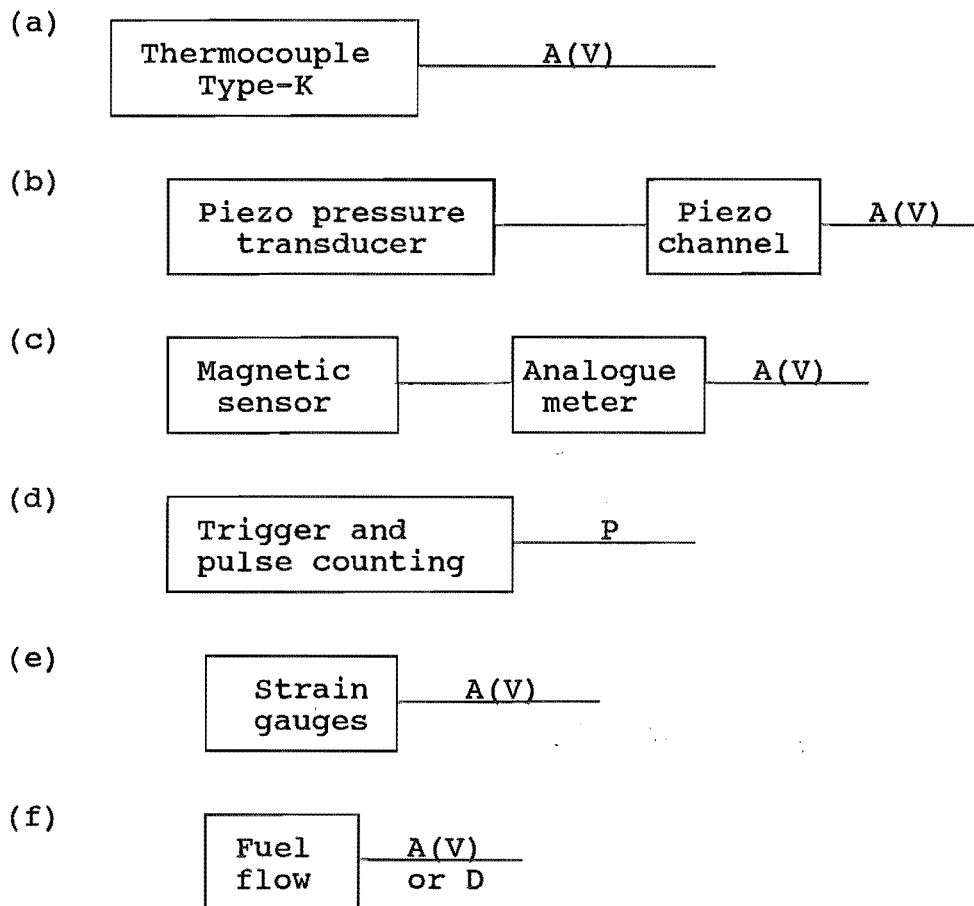


Figure 6.1 Data acquisition signal sketch map

Control signals can be summed up as follow:

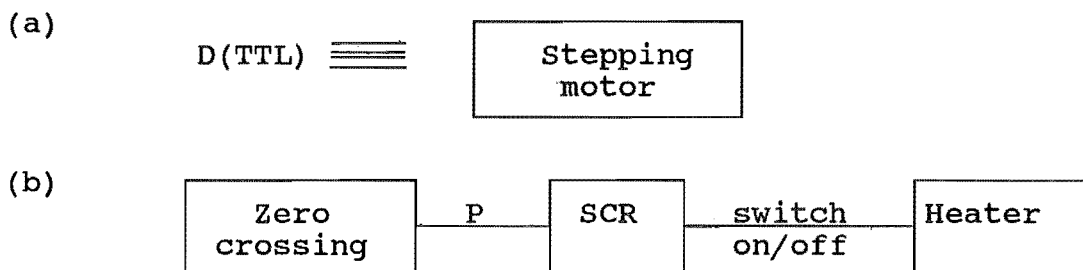


Figure 6.2 Control signal sketch map

II. DATA ACQUISITION AND CONTROL SYSTEM

A preliminary study has shown that it is possible to extend the fumigation technique to a direct injection multi-cylinder diesel engine. At the latter stage of this study, with the available computer, A/D converter and timer/counter board, a control system was developed which was based on the engine operating control parameters of fumigation and inlet air heating. This enables the current steady state operation to be extended to cover transient operating conditions and allows the system to be considered for future automotive use. Overall lay-out of the computer and its accessories is displayed in plate 6.1.



Plate 6.1 Overall lay-out of the computer and its accessories

An existing data acquisition and control unit from MetraByte corporation is suitable for data logging, process control and signal analysis etc. It comprises of a MetraByte DAS-8 data acquisition system, STA-08 screw terminal board and an EXP-16 expansion multiplexer/conditioner board. Therefore it is suitable for the requirements of this study.

From the baseline results, a wide range of fumigation and inlet air heating levels are necessary at certain engine speeds, torque and various operational temperatures. A control system can be conceived to take account of all of these factors and achieve optimal engine performance. The block diagram of the control system for the E6 engine appears in figure 6.3.

(1) Introduction to DAS-8 (48)

To better understand the control system, the following highlights the important functions of the DAS-8 board.

DAS-8 is an 8 channel, 12 bit successive approximation A/D converter with sample/hold. The full scale input of each channel is ± 5 volts with a resolution of 0.00244 volts. A/D conversion time is typically 25 microseconds (35 microseconds max.) and using the supplied software drive, throughput of up to 4000 sample/second are attainable when operating under BASIC. This fulfils the requirement of data collection for this study.

An 8254 programmable counter/timer provides periodic interrupts for the A/D converter and can additionally be used for even counting, pulse and waveform generation,

frequency, period and pulse width measurements.

There are three separate 16 bit down counters in the 8254. One of these is connected to a submultiple of the system clock and all I/O functions of the remaining two are accessible to the user. Input frequencies up to 2.5 Mhz can be handled by the 8254.

The 7 bits of TTL digital I/O provided are composed of one output port of 4 bits and one input port of 3 bits. Each output will handle 5 standard TTL loads. All I/O is accessed via a call statement: CALL DAS 8 (MD%, DIO%, FLAG%) when programming. There are 18 modes of operation (mode 0 to mode 17) available to the programmer.

Other features include: attached software (graphics, calibration, linearization, installation, I/O drive), precision 10 volt reference output, ± 12 and $+ 5$ volt power from the IBM PC/XT computer, programmable scan rate and foreground/background operation. A block diagram of DAS-8 is shown in figure 6.4.

The DAS-8 usually operates with its accessories. One of these is a model STA-08 screw terminal connector board, which allows a variety of connections to be easily made to the DAS-8. Its digital I/O port lines are monitored by L.E.D.'s.

Another accessory to be used is the model EXP-16 expansion multiplexer/conditioner, which multiplexes 16 differential inputs to a single output suitable for connection to one input channel of DAS-8. Up to 8 EXP-16 boards can be cascaded together on a single DAS-8 for a total 128 channels of analogue inputs. Selectable gains

between 1 and 1000 are available and a temperature sensor on the board allows the monitoring of cold junction temperature, thus the EXP-16 is ideally suitable for measuring thermocouple inputs. Its block diagram appears in figure 6.5.

(2) How to Obtain Acquisition Data

(a) Temperature Signals

The same thermocouple, unshielded chromel-alumel, is used in the data acquisition system. Each thermocouple is connected to the a Hi and Lo end of a corresponding channel on the EXP-16 board.

- .CH0 -- Ambient air temperature
- .CH1 -- Manifold temperature
- .Ch2 -- Exhaust temperature
- .CH3 -- Water temperature
- .CH4 -- Oil temperature
- .CH5 -- Heated inlet air temperature

A dip switch on the EXP-16 allows the user to select a wide variety of gains between .5 and 1000. The suitable gain is 50 for K-type thermocouple. This high performance instrumentation amplifier is suitable for use with strain gauges and other low level transducers.

The EXP-16 provides all the hardware needed for cold-junction compensation. The compensation signal can be sent to any of EXP-16's output lines by placing the jumper on counter J3 in the proper location.

The input channel of EXP-16 is chosen using "mode-14", then the output channels will remain fixed, and any subsequent DAS-8 converter routines will be performed on

that channel. To read other channels the "mode-14" routine should be called again and a new input channel selected. In this study, 8 input channels are scanned, and the A/D conversion are performed in "mode-4". All the data will be stored in an array E%(SUB%). Detail can be found at step 6 in the attached programme (Appendix B): EXP-16 measurements.

(b) Pressure Signal

When the DAS-8 data acquisition system is used to display the accurate pressure versus crank angle trace, it should also display signals of trigger and sampling rate. The trigger signal is picked up by a light sensor through a slot on the edge of a trigger disk, which is mounted on the shaft of the injection pump. It triggers the DAS-8 every complete engine cycle. A timing signal disk is marked every 5° around its circumference, the signal is detected by a magnetic pick up unit. The DAS-8 can interpret this signal and collect the information to plot a pressure diagram with divisions every 5° of crank angle, this can also be redisplayed as a pressure versus volume diagrams by the means of suitable software.

(c) Speed Signal

The generation of this signal was discussed in Chapter IV. This analogue signal is fed to CH(4) of the STA-08 screw terminal board. After conversion by "mode-5", the speed signal is recorded in array B%(4).

(d) Engine Torque

A load beam was connected to the dynameter casing on which a strain gauge was fixed on the top and bottom

surfaces. As the dynamometer casing tries to rotate under the engine torque, a bending stress in the load beam will be induced, which is sensed by the strain gauges and an electrical output is achieved.

The strain gauge has a +5 volt power supply from the EXP-16 board, which comes originally from the IBM PC. Its output is fed to output channel 7. The effect of temperature on the strain gauge is reduced through the cold junction compensation. Because it is a low level transducer, it is amplified by the EXP-16 and its processing is the same as that for the temperature signals -section (a).

(e) Fuel Flow Rate

At the last phase of this study, a fuel flow rate transducer (PIERBURG type 116-H) powered by +12 v DC, was available. It has both digital and analogue output. Analogue signal output was fed to the ch(4) of the STA-08. For calibration refer to Appendix A.

III. ESTABLISHMENT OF CONTROL

(1) Heater Control

Alcohol fuels have a high latent heat of vaporisation. A heater is therefore needed to raise the inlet air temperature to accelerate the vaporisation of this fuel when they are mixed in the carburettor and intake manifold. According to the baseline results, the temperature at the inlet manifold plays a much more important role in ensuring good combustion than the inlet air temperature. The reason for this is obvious, the

manifold temperature varies with the change of fumigation level in the carburettor whilst the inlet air temperature is kept constant. This results in a low mixture temperature in manifold (T_m) even though the inlet air temperature (T_{in}) is very high when fumigated fuel reaches the maximum level, and high temperatures in the manifold when the fumigation level is a minimum. Uncontrolled heating can lead to deterioration of combustion and even knocking, also too high a temperature will reduce the volumetric efficiency. Hence accurate control of the manifold temperature is essential and should be set to obtain a satisfactory mixture temperature. Figure 6.6 displays the control concept designed to realise this objective.

The zero crossing unit has a transformer 230 V AC/ 5 V DC. The sine wave is changed into pulses in this unit. These pulses go to counter 2 of the DAS-8 board. When there is a processing command, the counter counts down from a loaded number. When counting is zero, a pulse goes to the SCR unit, which works as a very high speed switch and switches on / off the power to the heater to heat up the inlet air to the required temperature.

The loaded number N was obtained through the tests and saved in the DATA subroutine. This number is decided by speed and torque of the engine. Later in the experiments, the computer picks up the signals of engine speed and torque, then ascertain the required temperature at the manifold (T_m), In the DAS-8, the "mode-10" is used to set the counter configuration, then the "mode-11" is used to load the counter. A pulse is sent to the SCR unit

to control the power supply to the heater after the counter counts to zero. If the real temperature in the manifold (T_r) is less than T_m , then the number will be reduced by an order $K*(T_r - T_m)$. Here K is a factor which can have a greater effort on the change rate of number N when a greater value is used. The smaller the number N , the more power that will go to the heater. This process lasts until the manifold temperature reaches the required T_m . When $T_r > T_m$, the number N will be increased thus T_r will decrease and finally close to T_m .

(2) Stepping-motor Control (49)

A Philips model ID35 stepping motor with a step angle of 7.5° was used to control the fumigation level. It was mounted on the shaft of the jet carburettor and controlled the fumigated fuel flow by turning the shaft.

The stepping motor was manoeuvred by a control board (plate 6.1), on which up to three stepping motors could be controlled. Both the stepping motor and the control board were powered by DC 5 volt. The board was linked to the output port of the computer, which normally is used for the printer. The stepping motor will rotate at a step rate of 7.5° each time it is activated. This is accurate enough for this control purpose.

Once the DAS-8 completes the EXP-16 and STA-08 measurements, and engine speed and load are defined, the software will pick up the required turning step of the stepping motor from the data subroutine. A comparison between the current position and the required end position of the stepping motor is made and the stepping motor will

then rotate to the required position. This will provide the required fumigation level.

Three subroutines are designed to conduct the function of turning the stepping motor for required fumigation control and setting the stepping motor to the start position if necessary.

IV. CONTROL PROGRAM "MODE-T"

The control programme "MODE-T" was defined by the baseline testing to operate the engine at an optimal level. It is based on the fact that the E6 engine was shown to be able to operate on alcohol fuels with the fumigation under accurate control. All the necessary data was collected in baseline testing before the programme was started.

Figure 6.7 is the block diagram of the control program "MODE-T".

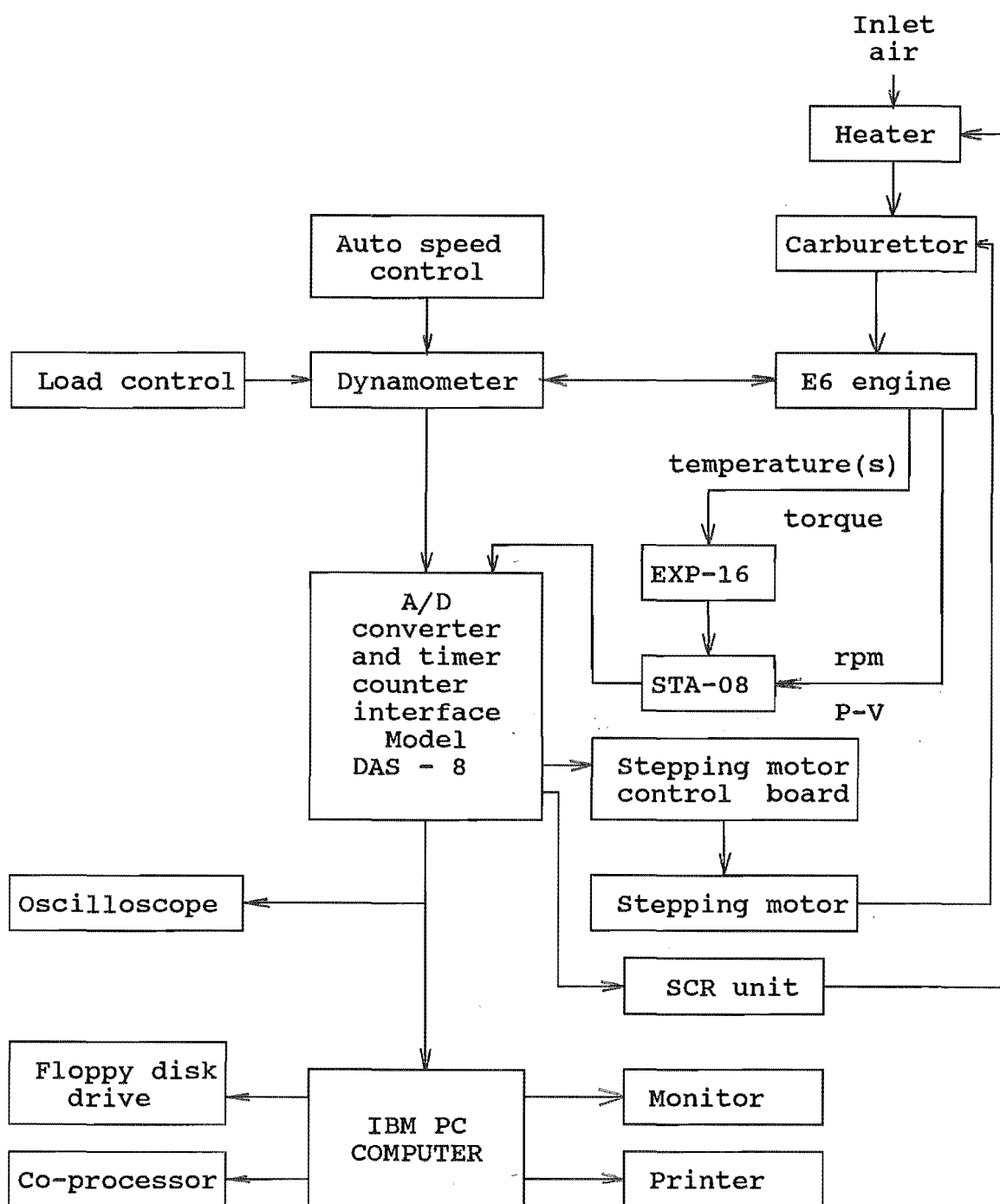


Figure 6.3 Block diagram of the control system

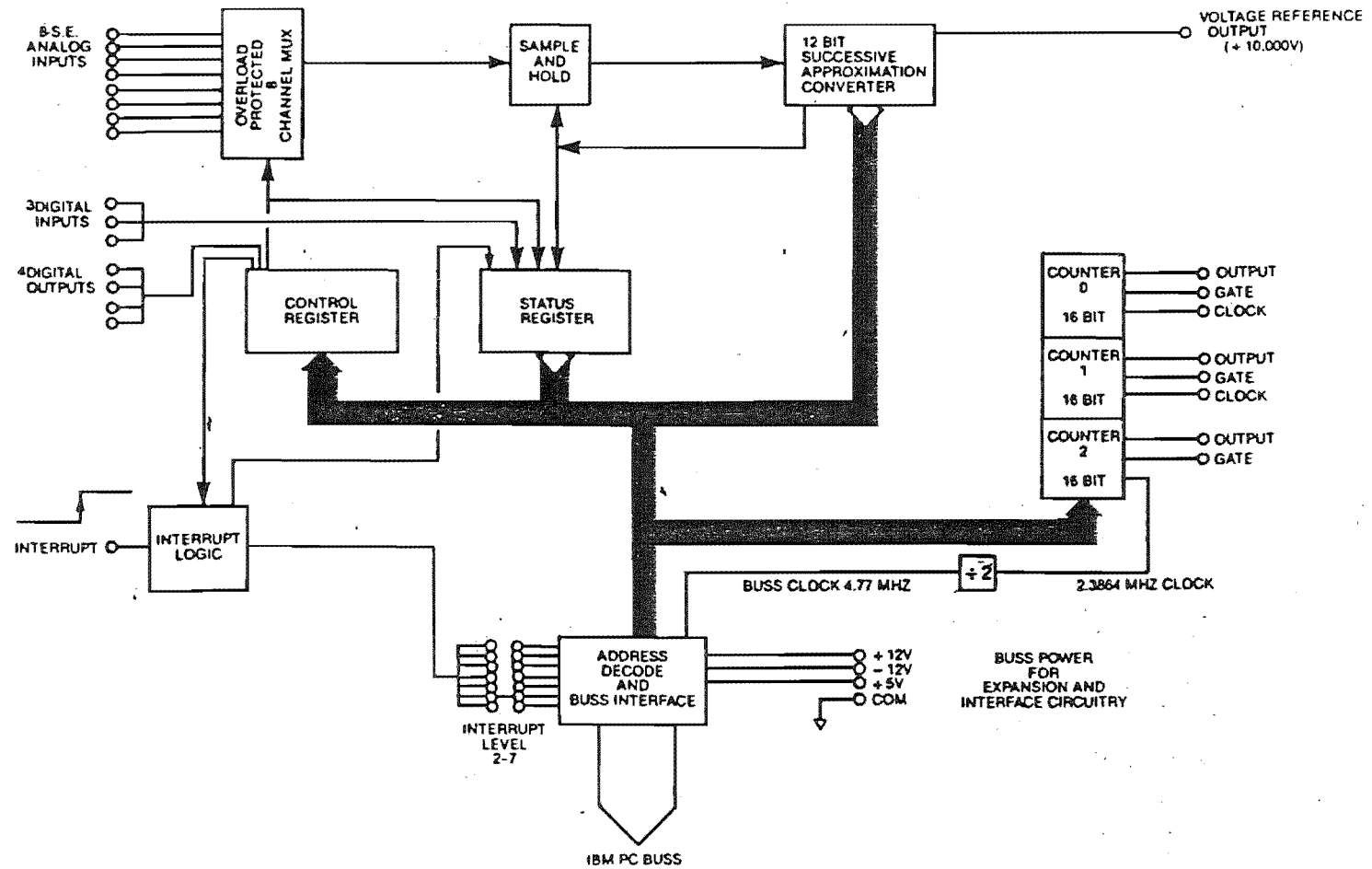


FIGURE 6.4 BLOCK DIAGRAM OF DASH - 8

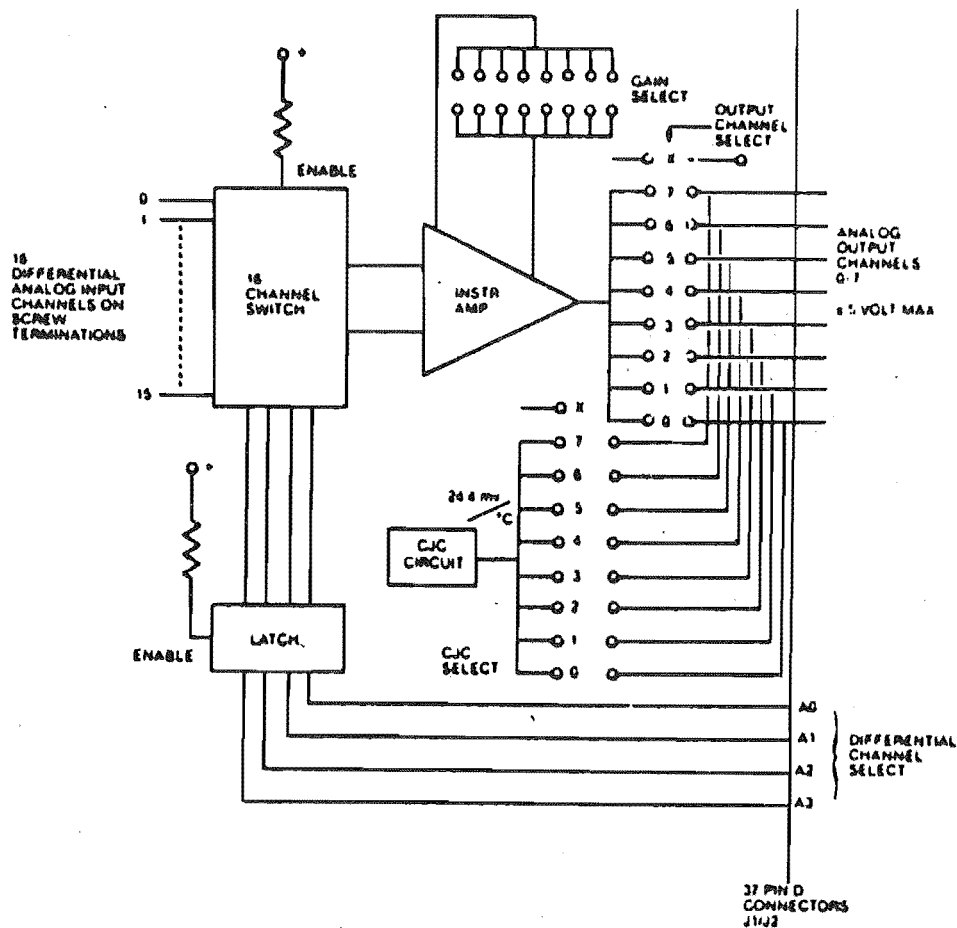


Figure 6.5 Block diagram of EXP-16

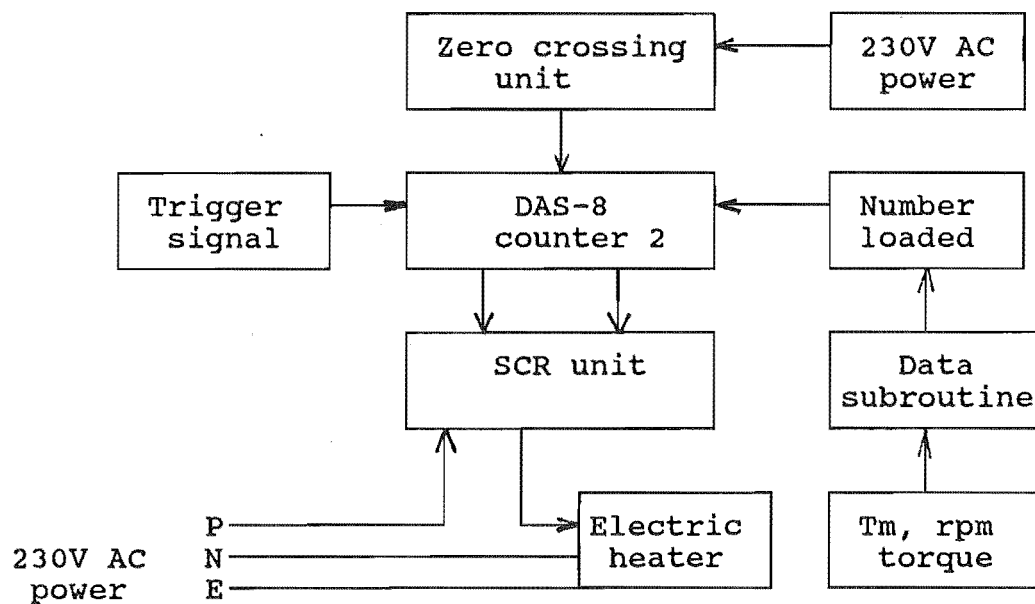
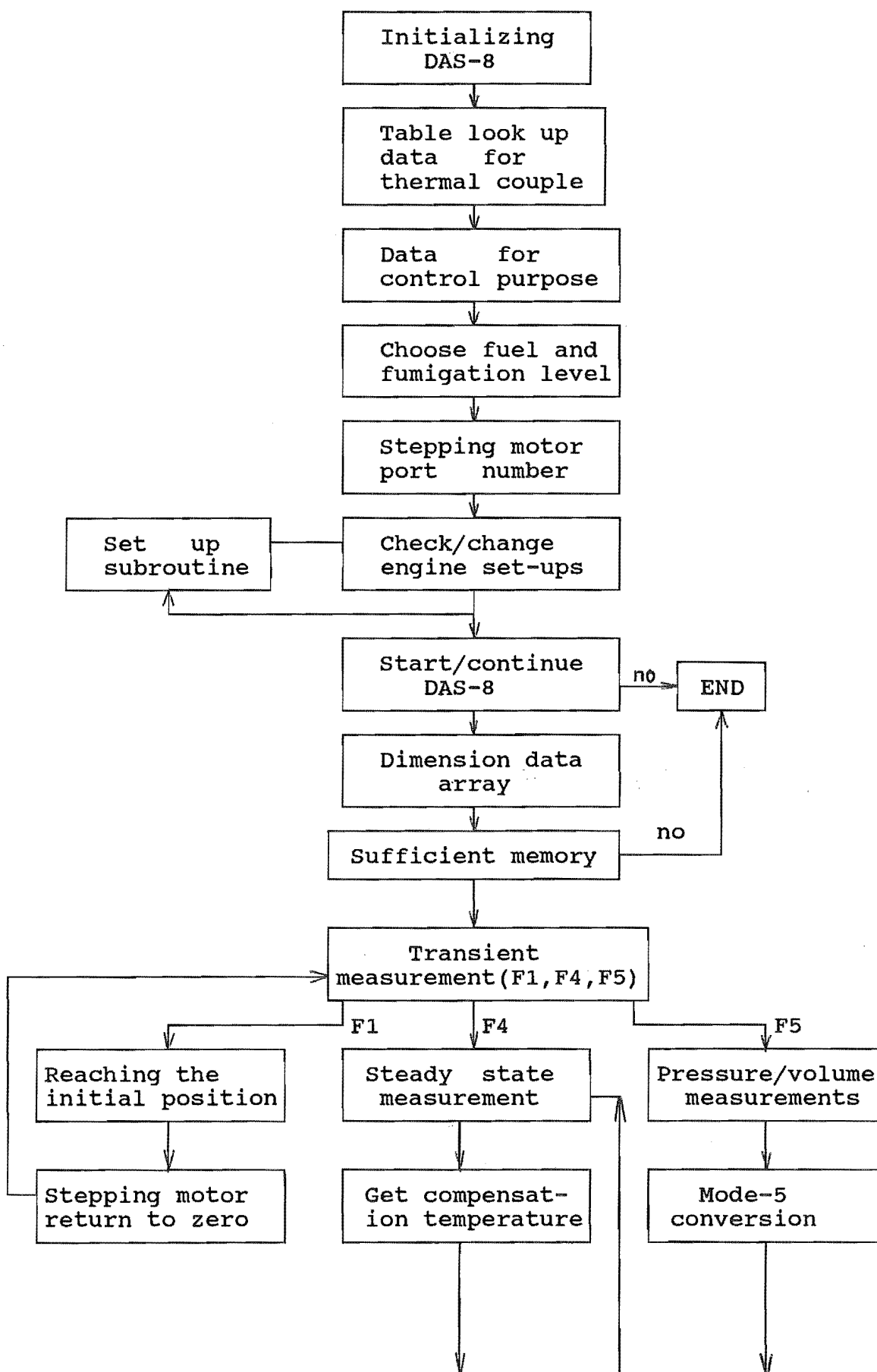


Figure 6.6 Heater control block diagram



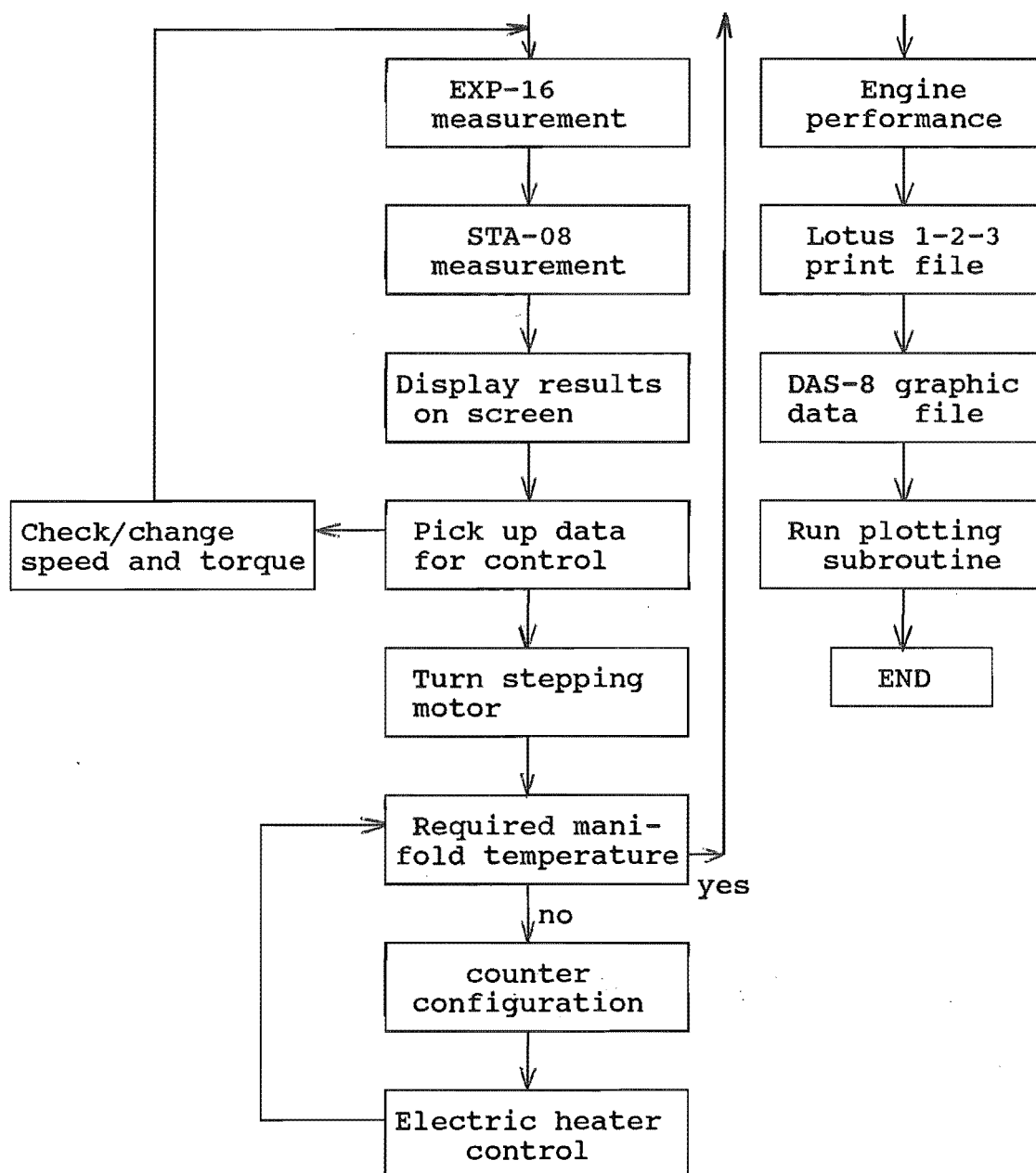


Figure 6.7 Block diagram of the control program "MODE-T"

CHAPTER VII

RESULTS FROM OPERATION OF CONTROL SYSTEM AND DISCUSSION

Described in chapter six was the study, design and building of a computer based data acquisition and control system. All of the necessary hardware was designed, built and tested as well as software development. This chapter will summarise the results obtained from use of this system and detailed discussions will be presented.

Refinement of the control system was carried out whilst the tests were conducted. For the E6 engine operating procedure refer to Appendix A. An explanation of the control programme "MODE-T" can be found in Appendix B.

I. SUMMARY OF TEST RESULTS

Test results obtained from operation of the engine on alcohol fuels are presented in figures 7.1 to 7.6. The computer programme "MODE-T" was run in the programmed pattern, i.e. at a predetermined speed, the engine ran at maximum or minimum fumigation level with optimal heating control as load changed according to the user's requirements.

(1) Methanol Operation

Figure 7.1 shows the engine performance on methanol. These results represent the engine performance parameters at two extreme fumigation conditions with controlled temperature of the air/alcohol mixture in the manifold.

Engine performance diagrams clearly show the

difference between the condition at maximum and minimum fumigation levels. The engine efficiency obtains its maximum under the maximum fumigation levels. Engine output is comparable to that achieved by diesel operation. Running conditions were satisfactory and reached their expected objective.

Figure 7.3 presents the fumigation changes versus brake power, which is associated with figure 7.1. The curves on the graph show the significant variation of fumigation levels, and this implies that the engine can be operated over a very wide range of conditions using the fumigation method.

Meanwhile, corresponding engine manifold temperatures (figure 7.5) were sustained in a range which could easily be achieved by a heat exchanger unit using as a heat source either the coolant or exhaust gas. It will be noted, however, that the corresponding heated air temperatures vary over a comparatively wider range.

(2) Ethanol Operation

In a similar way to methanol, engine operation on ethanol achieved comparable test results, which are shown in figure 7.2, 7.4, 7.6.

The engine performance curves are similar to those in figure 7.1. But the curves obtained at 2000 rpm show less variation between the extreme fumigation levels, this reflects less flexibility for engine operation.

Figure 7.4 displays the associated fumigation levels used for the results in figure 7.2. It also shows the

possible fumigation range at 2000 rpm is smaller than that at lower engine speeds, e.g. 1500rpm.

A related temperature diagram is presented in figure 7.6. Again the manifold temperatures fall into a range which could be achieved by means of a low temperature heat exchange unit. This paves the way for the establishment of an economic heat exchanger for future automotive use.

Therefore this control system shows great suitability to both methanol and ethanol fuels and achieves the basic target of this study.

II. COMPARISON AND DISCUSSION

(1) Comparison to the baseline test results

(a) Methanol Operation

The experimental set-up for results displayed in figure 7.1 is based on the results displayed in figure 5.3 and its associated set-up in figures 5.5, and 5.9. Because of their similar fumigation characteristics (figures 5.5 and 7.3) and controlled manifold temperature (figures 5.9 and 7.5), these performance diagrams have a high approximation tendency. The following discussion therefore concentrates on the differences between the results obtained by the control system and those in the baseline tests.

(i) **The Maximum Power Output** - The maximum power output has not reached the same value obtained in the baseline tests. This is to avoid the possibility of uncontrolled knocking that easily occurs at the extreme fumigation condition if there is not a strictly controlled

manifold temperature. Such knocking could severely damage the test engine. To avoid these unfavourable critical conditions, maximum engine output was restricted by the programme "MODE-T" with the arrangement of suitable control data in the data subroutine.

(ii) **Fumigation Effect** - In figure 7.1, generally higher thermal efficiencies were noted when compared to the baseline test, figure 5.3. These are due to the higher minimum fumigations level that were used instead of the corresponding baseline test levels displayed in figure 5.5. This higher level was to avoid the onset of misfire under the control condition. Higher minimum fumigation levels will lead to the higher thermal efficiency, this has been discussed in chapter five (refer to figure 5.19).

(iii) **Temperature Effect** - In chapter five, the manifold temperature, T_m , is noticed to be more important than the inlet temperature, T_{in} , for the effective engine control. Hence T_m is controlled according to a T_m data subroutine in programme "MODE-T". The relationship in the variation of manifold temperature with minimum and maximum levels of fumigation is similar to the baseline tests and when using the control system; i.e. minimum fumigation levels require greater manifold temperature to achieve satisfactory combustion. However, there are some differences in their exact values. This is shown in figures 7.5 and 5.9. Comparable engine performance curves are presented in figures 7.1 and 5.3.

From the diagram of brake power and thermal efficiency versus inlet air temperature (figure 5.21), the

small temperature change will have little effect on the brake power and thermal efficiency, say T_{in} changes from 40 °C to 80 °C. The thermal efficiency only has a slight increase of 1.5%. Therefore the difference in temperature has a lesser impact on engine performance whilst the fumigation level plays a more dominant role (figure 5.19).

(b) Ethanol Operation

Because of the similar characteristics of the alcohol fuels, a close comparison can be made between these results with ethanol and the early methanol results.

(i) **Maximum Power Output** - The E6 engine maximum power output has not achieved that obtained in the baseline tests, again because it was artificially restricted. This means insufficient levels of fumigated ethanol were used to compensate for the inadequate supply of injected fuel, hence less power output was obtained. The reason for the artificial restriction is the same as that mentioned above with regard to methanol.

(ii) **Fumigation Effect** - From the figures 5.4 and 7.2, the latter displays a generally higher level of thermal efficiency especially at higher load under the minimum fumigation level. This is because similar measures, as for methanol, were used here to increase the minimum fumigation level artificially in order to avoid the onset of misfire. These factors are obvious on the curves with legend i.f. (minimum fumigation) in figure 7.4.

A narrower fumigation level range means a slightly less flexible engine operating performance as shown in figure 7.2 when compared to figure 5.4. The change of

fumigation level was a paramount factor in the thermal efficiency change.

(iii) **Temperature Effect** - These results can be referred to in figures 7.6. It was noticed that the engine performance curve at 2000 rpm under minimum fumigation in figure 7.2 was very close to that under maximum fumigation, and even had a higher thermal efficiency at the lower load. That is due to the different trend in curves of T_m at the lower load (figure 7.6) when compared to that in figure 5.10. The corresponding manifold temperature, approximately 48 °C, is comparatively higher than that of 23°C under maximum fumigation as shown in figure 7.6. This higher manifold temperature makes an important contribution to the higher thermal efficiency at 2000rpm under minimum fumigation in figure 7.2.

(2) CONCLUSION

From the above comparisons and discussion, the following conclusions can be drawn:

- (a) The DAS-8 data acquisition and control system has demonstrated its practical use for the control of the fumigation system and is suitable for both methanol and ethanol fuels.
- (b) Data acquisition processing is prompt and quick, response from the developed software "MODE-T" was satisfactory for the control purpose.
- (c) The fumigation level is controlled effectively with programmed heating

assistance for vaporisation of alcohol fuels. A wide range of engine operation is available through the organisation of control data subroutines.

- (d) Both methanol and ethanol showed great adaptability to the developed fumigation system and achieved the target of near 100% replacement of diesel fuel except for 2% water addition to avoid corrosion problem and 2% castor oil added to the alcohol to improve its lubricating ability.
- (e) Comparable engine performance was achieved by the E6 engine operation on methanol and ethanol when compared to diesel operation.
- (f) The comprehensive results can be used as a basis for developing a system for automotive use.

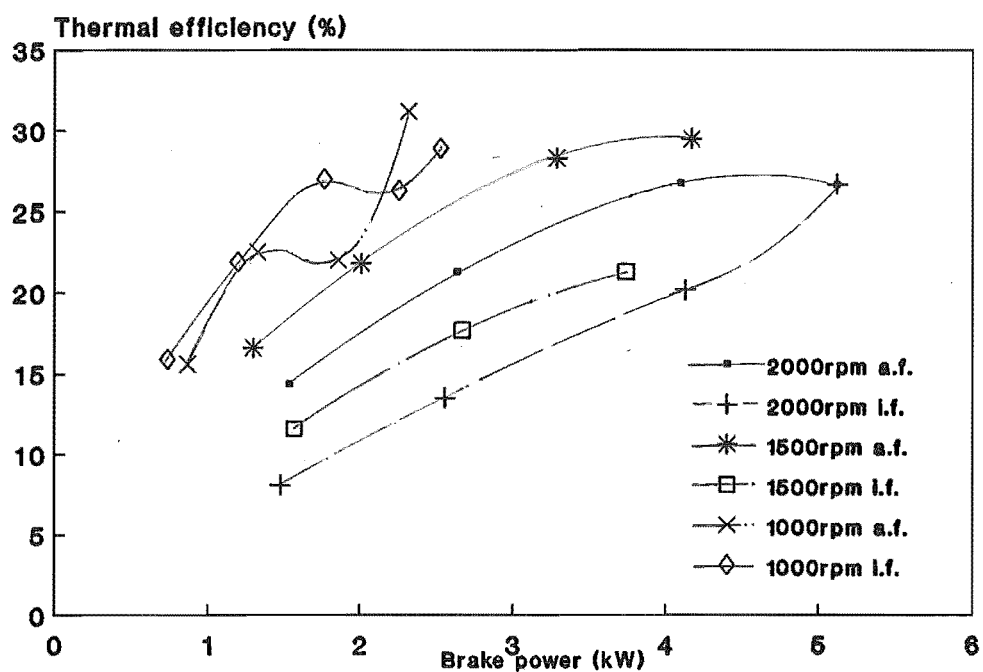


Figure 7.1 Engine performance on methanol

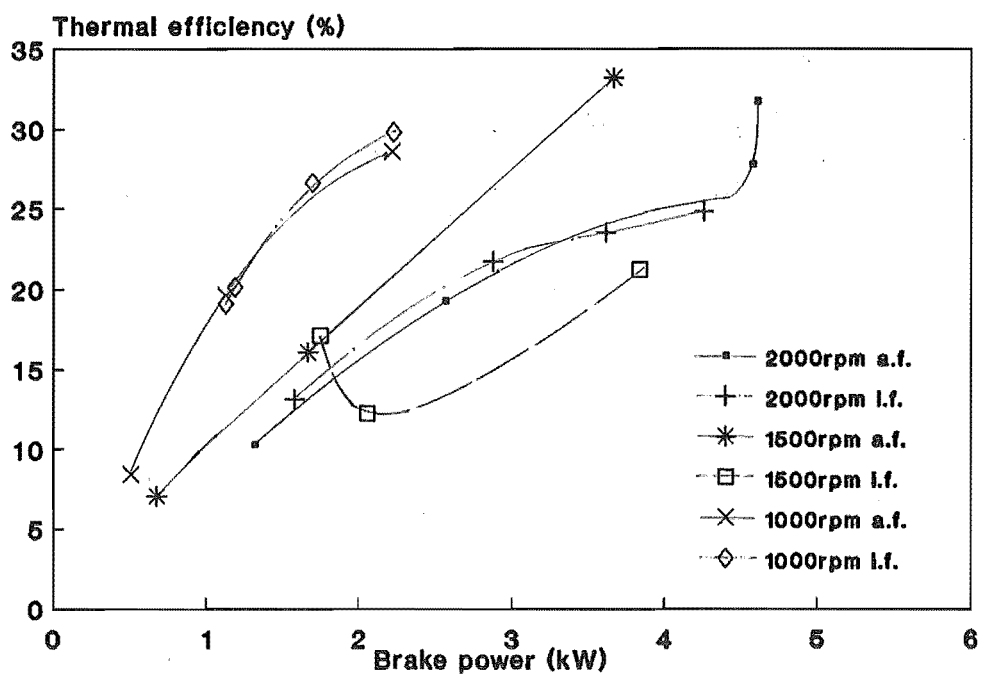


Figure 7.2 Engine performance on ethanol

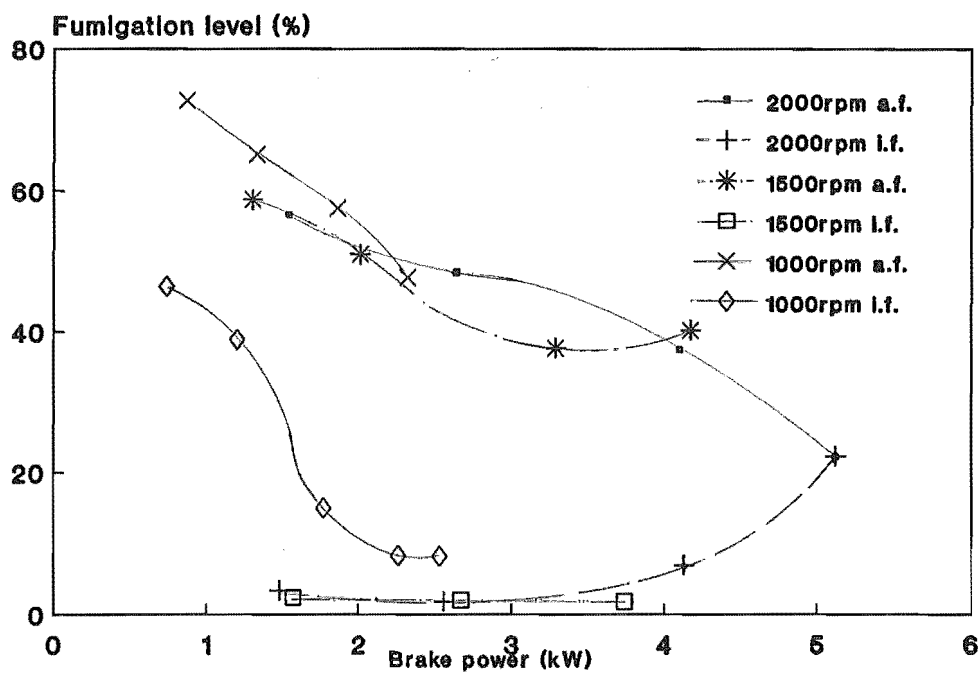


Figure 7.3 Fumigation vs brake power (methanol)

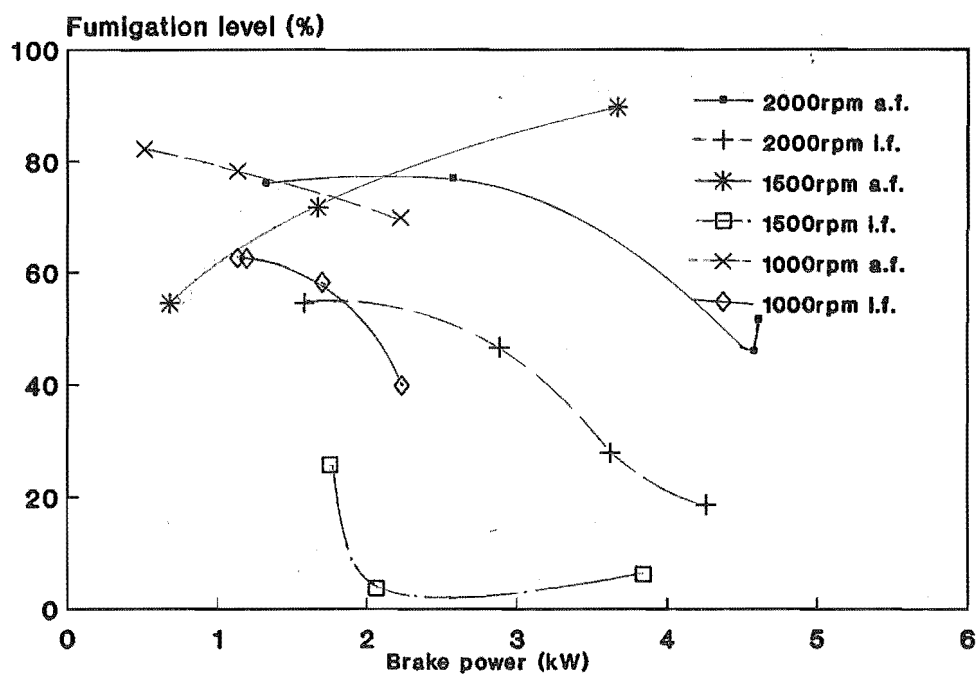


Figure 7.4 Fumigation vs brake power (ethanol)

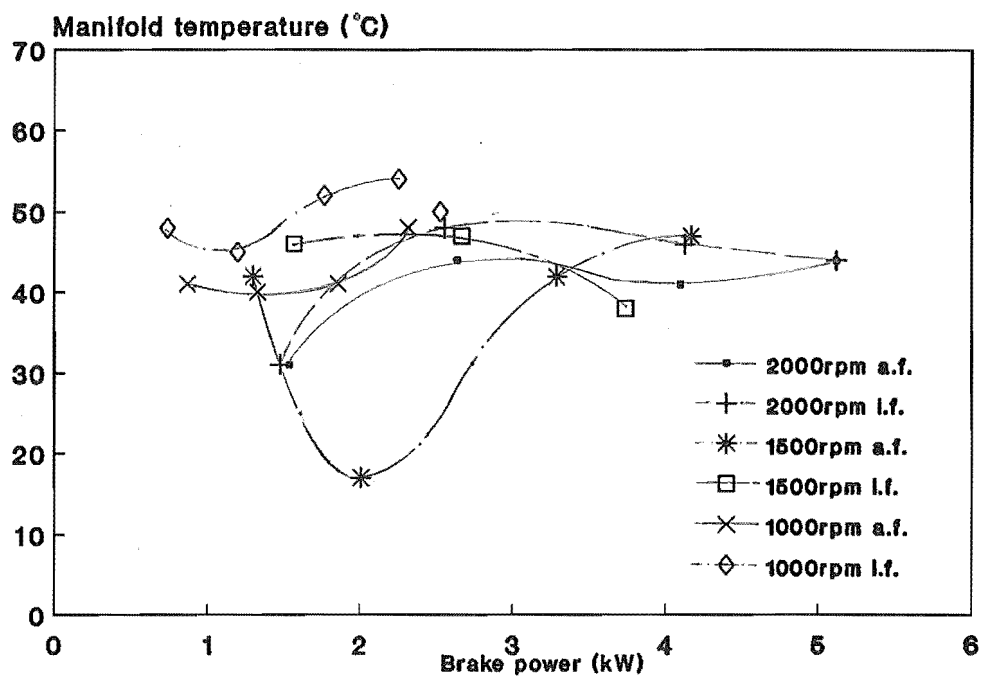


Figure 7.5 Manifold temperature vs brake power (methanol)

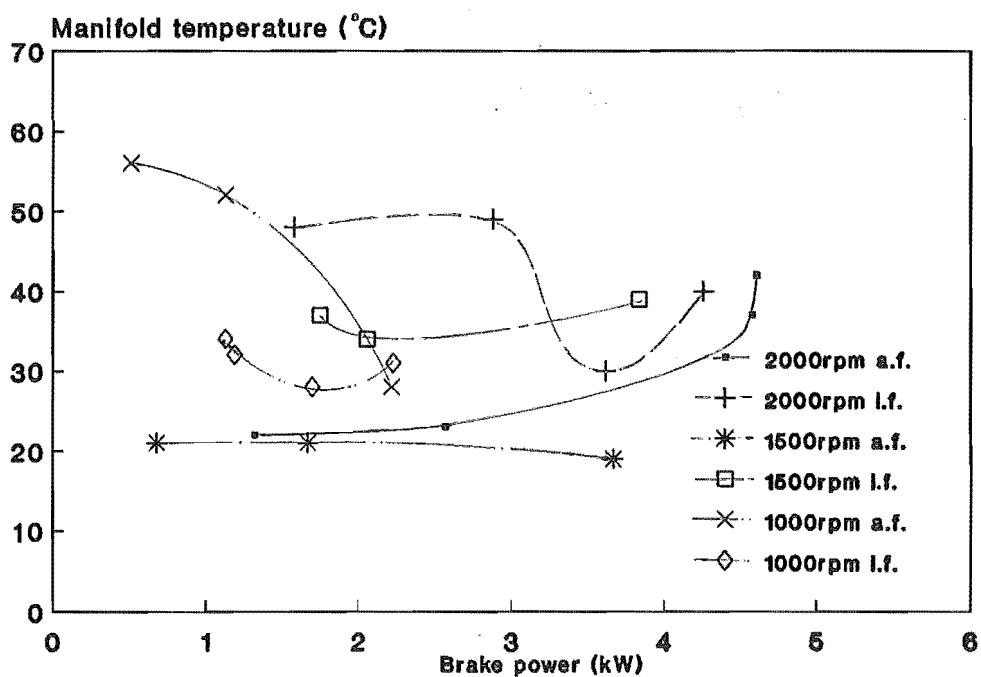


Figure 7.6 Manifold temperature vs brake power (ethanol)

CHAPTER VIII

CONCLUSIONS AND RECOMMENDATIONSI. INTRODUCTION

The major objective of this study was to explore a simple and inexpensive technique to enable a research diesel engine to operate on the alcohol fuels. Amongst the methods previously investigated by other researchers, successful solutions appeared to be both costly and complex. Fumigation, however, appeared to provide an inexpensive, attractive solution to the problem and offered potential for the practical automotive use. This study therefore focused on this method. Subsequent test results of a fumigated system proved the assumptions as to its suitability to be correct. A small single cylinder research diesel engine can operate in a satisfactory manner on near pure alcohol fuels.

Although fumigation method can be applied to direct engine operation on near 100% alcohol fuels, it requires careful control of the fumigation levels and auxiliary mixture heating levels to provide for a wide engine operation range and optimal engine performance. Therefore a control system had to be built to realize these parameters. The following will highlight the results achieved with the above objective.

II. SUMMARY OF TEST RESULTS

(1) Baseline Results

A diesel engine can run on nearly 100% methanol or ethanol except for the addition of 2% distilled water and 2% castor oil to the alcohol.

Engine performance when operating on alcohol fuels is comparable to that obtained by diesel fuel. The power outputs are even higher at lower engine speed with similar engine thermal efficiency.

Fumigation plays an important role with regard to engine performance. It promotes not only the auto-ignition of low cetane number injected alcohol fuel, which is deemed to be an obstacle for diesel engine operation, but also improves engine thermal efficiency at higher levels of fumigation. However, the maximum level is limited by the occurrence of knocking.

Auxiliary heating of inlet air is also necessary to help vaporize the fumigated alcohol. A hotter fumigated mixture was shown to be beneficial in improving flame speed and fuel ignitability, hence achieving higher engine thermal efficiency and smoother operation conditions. However too high the heating level will lead to a reduction in volumetric efficiency of the engine.

(2) Results of Control System Operation

The DAS-8 data acquisition and control system can control the engine operation precisely within the programmed range.

The trend in the graphs of engine performance, fumigation level and manifold temperature obtained by the control system is similar to that of the baseline tests.

This is expected since a matrix of data from the baseline tests is employed by the control program.

Some results are not identical to those of the baseline tests because higher minimum fumigation levels were used to avoid the onset of misfire and lower maximum levels used to prevent knock formation. Different fumigation levels lead to a slight change in the temperature of the fumigated mixture (T_m).

In general, this control system reaches its objectives whilst at the same time allowing the engine to operate within safe working limit.

III. CONCLUSIONS

- (1) An inexpensive conversion system developed on a single cylinder research engine enabled a diesel engine to operate on near 100 percent methanol or ethanol. These results have not been found in the published literature.
- (2) A fumigation method was used for this system. It involved fumigating heated inlet air with alcohol fuel, thus overcoming the technical difficulty of the extremely low auto-ignition capability of alcohol fuels.
- (3) Both sets of results, from the baseline tests and the computer control system, prove that the engine achieves satisfactory combustion characteristics and the performance is comparable to that obtained

on diesel fuel.

- (4) A DAS-8 data acquisition and control system precisely controlled the engine operating conditions. Such control is necessary, can easily be achieved and reduces the likelihood of any human errors.
- (5) These comprehensive results should encourage future investigations to develop the system for automotive use.

IV. RECOMMENDATION FOR FUTURE WORK

- (1) Further precise refinement work of the software is required to achieve a wider ethanol operating range.
- (2) A heat exchanger using as a heat source either the engine coolant or exhaust gas is required to be designed and built in order to reach the target of a low cost conversion kit.
- (3) Cold starting problems need to be addressed.
- (4) The exhaust emissions, especially NO_x and particulate matter, are usually lower with an engine operating on alcohol than that on diesel. Some reference information is presented in Appendix C. In this study NO_x and particulate matter were not measured because of the necessary instrumentation was not available. Hence emission studies need to be carried out.

- (5) Adaptation of the fumigation and control system for automotive use and testing using the chassis dynamometer should be investigated. Such a system could be developed based on the results presented in this study.

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APPENDIX A**ENGINE START-UP AND CALIBRATION PROCEDURE****A.1. ENGINE START-UP PROCEDURE****A.1.1. Starting Ricardo Engine**

1. Push **START** on rectifier.
2. Turn **RICARDO** switch on (on rectifier).
3. Turn on lights, fan and air conditioning and open window.
4. Fill fuel tanks.
5. Turn on fuel taps.
6. Check water level in column.
7. Zero manometer (air flow).
8. Set compression ratio.
9. Set diesel pump timing.
10. Check water and oil temperature controllers.
11. Turn on AC supply switch.
12. Check **STOP** button is released.
13. Push switches for **WATER, OIL PUMPS** and **OIL HEATER**.
14. Turn on the water heater (in coolant column).
15. Check oil pressure 2 bar.
16. Push **MOTORING** switch.
17. Turn **MODE** switch to **DIESEL**.
18. Set speed controller to 405 (1500 RPM).
19. Bring up start handle slowly until engine starts, then release.
20. Turn main load handle wheel off (anti-

clockwise).

21. Push **LOAD** button.
22. Push No.4 button on temperature switch for water temperature (70° - 75° C).
23. Turn main load handle wheel on to raise engine load.
24. Run the engine until oil and water temperatures nearly reach the set temperatures and then turn water heater off.
25. Turn the diesel fuel tap to the off position on indicator plate.
26. Watch the fuel level drop below the inlet tube, then loose the ventilating nut and let the air in until no fuel is visible in the fuel injection pump.
27. Turn the tap to alcohol position and tighten the ventilating nut.
28. Adjust the fumigation level by turning the indicator on carburettor.
29. Supply suitable electric power to the heater to warm up the inlet air until satisfactory audible combustion is obtained.

A.1.2. Stopping Ricardo Engine

1. Turn off the electric heater.
2. Wind back speed controller to Zero.
3. When engine is stationary, turn off **OIL, WATER PUMPS and HEATER.**
4. Turn off AC supply switch.
5. Turn off cooling water.

6. Turn off fuel taps and fumigation supply.
7. Turn engine to TDC compression stroke.
8. Turn off air conditioning, fan, lights and shut window.
9. Turn off rectifier.

A.2. CALIBRATION OF TRANSDUCERS

A.2.1. Calibration of Engine Speed

1. Start the engine according to the section A.1.
2. Turn on the PC/XT computer.
3. Run C:\cd\fred\mode-t.
4. Set the engine speed factor to 1 (rpm/V).
5. Sample with engine speed from 0 to 2000 rpm, record the output reading (Volt) of engine speed signals at 0, 500, 1000, 1250, 1500, 1750, 2000 rpm. Engine speed reading comes from the mechanical tachometer.
6. Plot the engine speed versus output voltage diagram (figure A1). (use LOTUS-123 worksheet to load the data; then input the lotus-123 worksheet to HG graphic package and get the change trend diagram, save it in a plotter file; use WORDPERFECT package to get the required size and print out).
7. Factor for MODE-T control program is 745.7 rpm/V.

A.2.2. Calibration of Strain Gauge

1. Start the engine according to the A.1.

2. Run C:\cd\fred\mode-t.
3. On the other side of dynamometer, increase the balance weight, record the corresponding output voltage of strain gauge.
4. Set the engine torque factor to 1 (Nm/Volt).
5. Engine torque is the product of weight times arm length.
6. Plot the diagram of engine torque versus output voltage of strain gauge (figure A2). Torque factor for MODE-T is 3.5 Nm/V.

A.2.3. Calibration of Fuel Flow-rate Transducer

1. Procedure 1 and 2 are the same as above.
2. Set the fuel flow-rate factor to be 1 (ml/s)/Volt.
3. Measure the output voltage of transducer.
4. Plot the diagram of fuel flow-rate versus output voltage (figure A3) and find found the factor (1.887 (ml/s)/V).

A.2.4. Calibration of Pressure transducer

Direct calibration of the piezo electric transducer using a dead weight tester proved difficult and inaccurate. Hence an indirect method was employed. The indicated power of an engine can be calculated by following expression:

$$I_p = \frac{P_{ind} V_{swept} N (rev/s)}{2 \times 10^{-2}}$$

Whilst there is another expression:

$$I_p = (2 \times 3.1416 \times N \times T_b + 2 \times 3.1416 \times N \times T_f) / 10^3 \text{ (kW)}$$

The T_b is the brake torque which can be measured by

test; T_f is the friction torque, this can be determined by means of a WILLANS LINE diagram. Therefore a pressure transducer factor ($K=P_{ind}/\text{volt}$) can be calculated after the measurement of voltage output corresponding to P_{ind} .

With the WILLANS LINE, T_f is 9 Nm (refer to figure a4). Calculated results are presented in the following table.

Run No	Torque Nm	Fuel flow kg/h	bp kW	I_{ind} kW	P_{ind} bar	V*36.33 Volts		Pressure Factor K bar/V	
						No.1	No.2		
1	22.61	.95	2.97	4.152	7.860	11.17	11.17	25.56	25.56
2	20.33	.81	2.67	3.852	7.293	10.36	10.07	25.58	26.31
3	15.99	.66	2.10	3.283	6.214	8.876	8.949	25.43	25.23
4	8.25	.43	1.08	2.266	4.289	5.740	6.238	27.15	24.98
5	3.77	.32	.49	1.677	3.175	4.193	4.417	27.51	27.81
6	2.94	.31	.39	1.568	2.969	3.480	4.469	30.99	24.13

Take the minimum and maximum results off and calculate the average value.

Pressure factor: $K=26.426$ (bar/V)

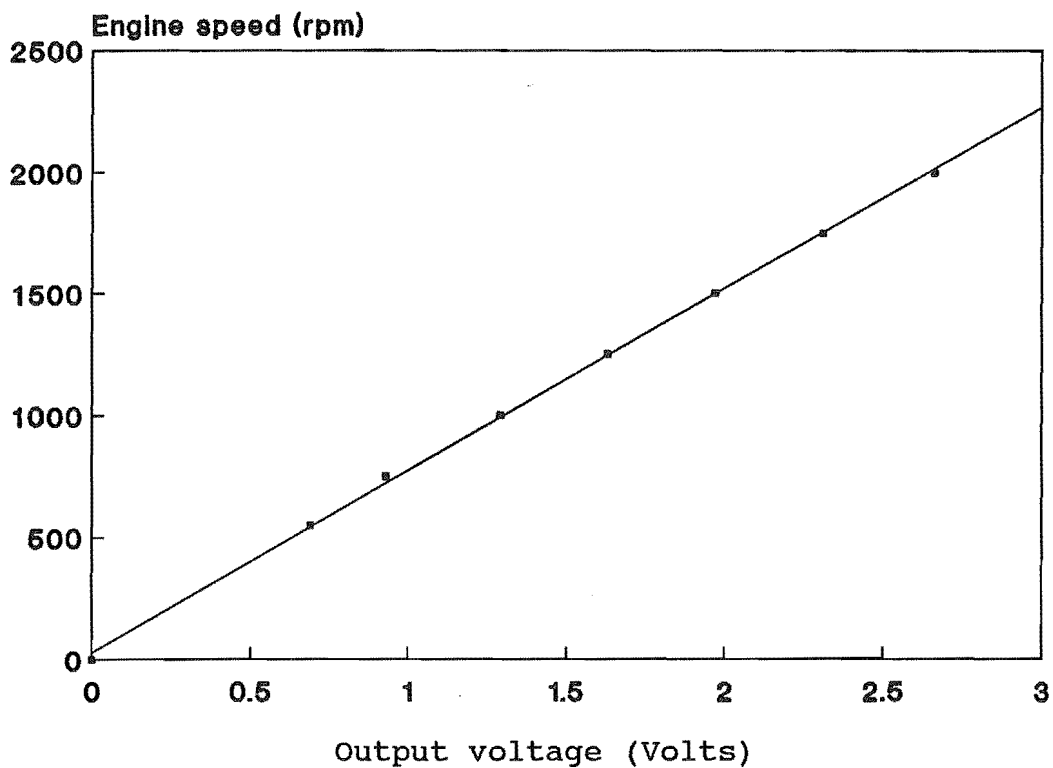


Figure A1 Calibration of engine speed measurement

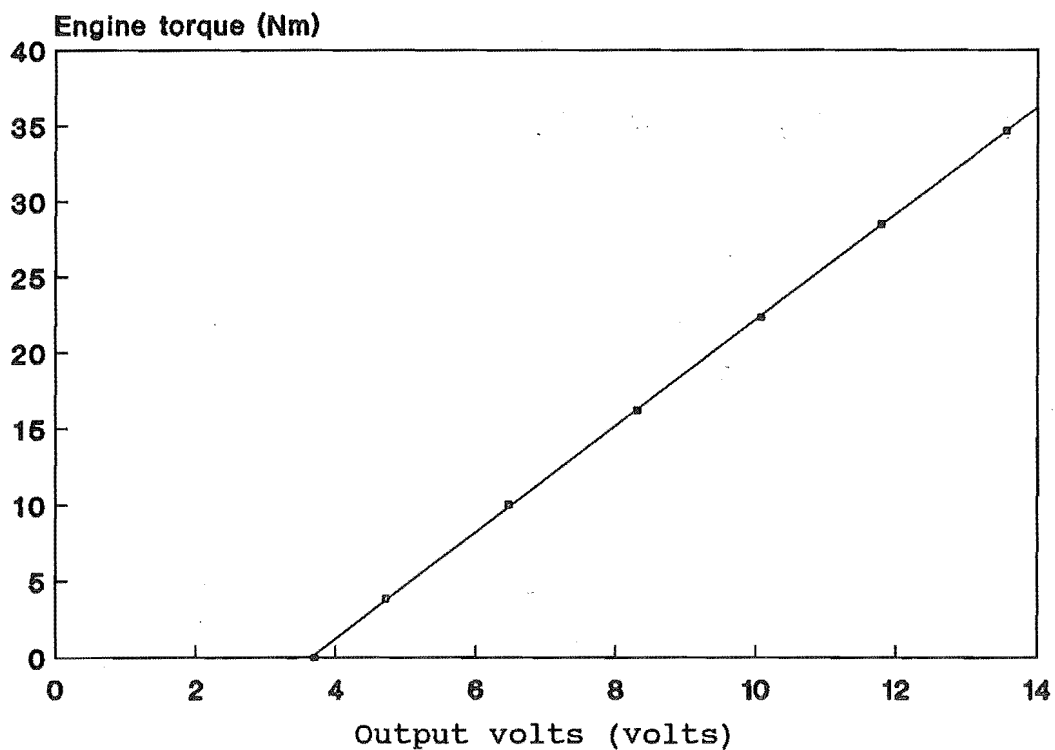


Figure A2 Calibration of strain gauge

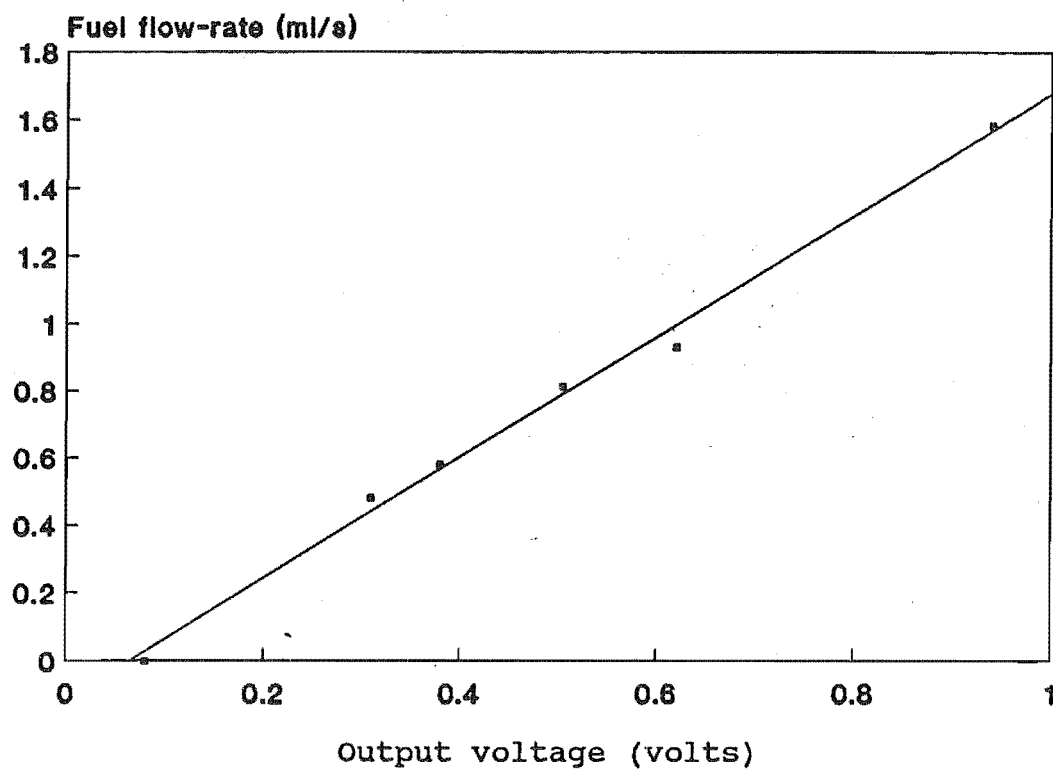


Figure A3 Calibration of fuel flow-rate transducer

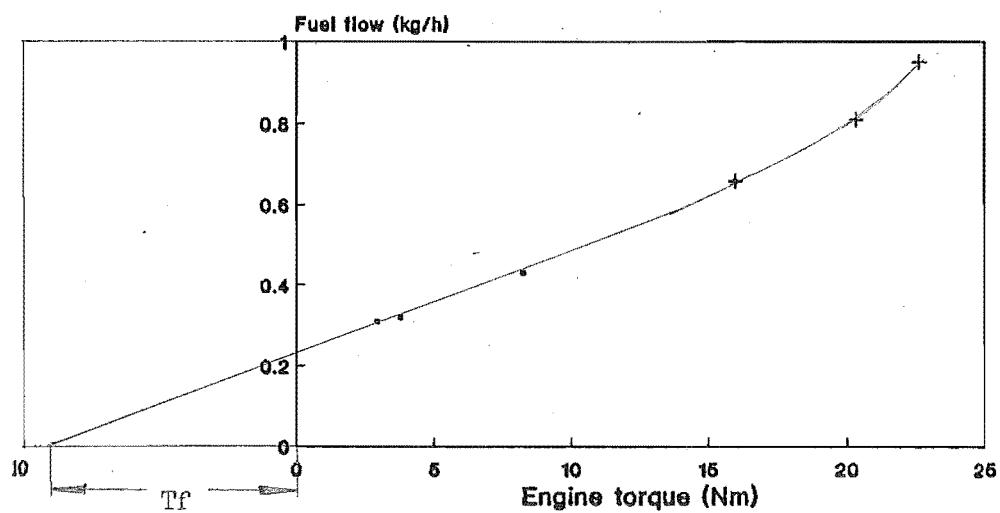


Figure A4 Willans line

APPENDIX B

"MODE-T" CONTROL PROGRAMME

```

100 '
150 !*****
200 '* A program for using DASH-8 and it's accessories (STA-8 & EXP-16) as a *
250 '* data aquisition device for the RICARDO-E6 engine operation on alcohols *
300 !*****
350 '
400 SCREEN 0,0,0 : KEY OFF : WIDTH 80
450 'Make initial screen announcement
500 CLS
550 PRINT" This program's purpose is to use the DASH-8 and it's accessories (the STA-8 "
600 PRINT"screw terminal accessory board and the EXP-16 expansion interface),as a data"
650 PRINT"aquisition system for the RICARDO-E6 engine's steady-state mode tests."
700 '-----
750 '---Step 1: Initialize DASH-8 with Mode 0 -----
800 '-----
850 CLEAR, 64000!,2500 'Contract BASIC workspace to 64K
900 'DEF SEG = 0 'Ready to read BASIC's working segment location
950 'SG = 256*PEEK(&H511) + PEEK(&H510)
1000 'SG = SG + 64000!/16 'Load location at end of workspace
1050 'DEF SEG = SG
1100 DEF SEG=&H5000
1150 BLOAD "DASH8.BIN",0 'Load driver routine
1200 DASH8 = 0 'Load offset = 0
1250 DIM D%(4) 'Data transfer array variable
1300 D%(0) = &H300 'I/O address of DASH-8 (change to suit)
1350 MD% = 0 'initialize mode
1400 FLAG% = 0 'declare error variable
1450 CALL DASH8 (MD%, D%(0), FLAG%)
1500 '----- Table lookup data for K type thermocouple -----
1550 'Run this subroutine only in the initialization section of your program
1600 'Number of points, voltage step interval (mV), starting voltage (mV)
1650 DATA 309 , .2 , -6.6
1700 READ NK, SIK, SVK
1750 'Temperature at -6.6mV, -6.4mV, -6.2mV etc.
1800 DATA -353.5,-249.3,-224.0,-207.6,-194.3,-182.8,-172.3,-162.8,-153.8,-145.4
1850 DATA -137.3,-129.6,-122.3,-115.2,-108.3,-101.6, -95.1, -88.7, -82.5, -76.4
1900 DATA -70.4, -64.6, -58.8, -53.1, -47.5, -42.0, -36.6, -31.2, -25.9, -20.6
1950 DATA -15.4, -10.2, -5.1, -0.0, 5.0, 10.1, 15.1, 20.0, 25.0, 29.9
2000 DATA 34.8, 39.7, 44.6, 49.5, 54.3, 59.1, 64.0, 68.8, 73.6, 78.4
2050 DATA 83.2, 88.0, 92.9, 97.7, 102.5, 107.4, 112.2, 117.1, 122.0, 126.9
2100 DATA 131.8, 136.7, 141.7, 146.6, 151.6, 156.5, 161.5, 166.5, 171.5, 176.5
2150 DATA 181.6, 186.6, 191.6, 196.6, 201.6, 206.6, 211.6, 216.6, 221.5, 226.5
2200 DATA 231.5, 236.4, 241.4, 246.3, 251.2, 256.1, 261.0, 265.9, 270.8, 275.6
2250 DATA 280.5, 285.3, 290.2, 295.0, 299.8, 304.6, 309.4, 314.3, 319.1, 323.9
2300 DATA 328.7, 333.4, 338.2, 343.0, 347.8, 352.6, 357.3, 362.1, 366.9, 371.6
2350 DATA 376.4, 381.1, 385.9, 390.6, 395.4, 400.1, 404.8, 409.6, 414.3, 419.0
2400 DATA 423.8, 428.5, 433.2, 437.9, 442.6, 447.3, 452.0, 456.8, 461.5, 466.2
2450 DATA 470.9, 475.6, 480.3, 485.0, 489.7, 494.4, 499.1, 503.8, 508.5, 513.1
2500 DATA 517.8, 522.5, 527.2, 531.9, 536.6, 541.3, 546.0, 550.7, 555.4, 560.0
2550 DATA 564.7, 569.4, 574.1, 578.8, 583.5, 588.2, 592.9, 597.6, 602.3, 607.0
2600 DATA 611.7, 616.4, 621.2, 625.9, 630.6, 635.3, 640.0, 644.8, 649.5, 654.2
2650 DATA 658.9, 663.7, 668.4, 673.2, 677.9, 682.7, 687.4, 692.2, 696.9, 701.7
2700 DATA 706.5, 711.3, 716.1, 720.8, 725.6, 730.4, 735.2, 740.0, 744.8, 749.7
2750 DATA 754.5, 759.3, 764.1, 769.0, 773.8, 778.7, 783.5, 788.4, 793.3, 798.1
2800 DATA 803.0, 807.9, 812.8, 817.7, 822.6, 827.5, 832.4, 837.3, 842.2, 847.2
2850 DATA 852.1, 857.1, 862.0, 867.0, 872.0, 876.9, 881.9, 886.9, 891.9, 896.9

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2900 DATA 901.9, 906.9, 911.9, 916.9, 922.0, 927.0, 932.0, 937.1, 942.2, 947.2
2950 DATA 952.3, 957.4, 962.5, 967.6, 972.7, 977.8, 982.9, 988.0, 993.1, 998.2
3000 DATA 1003.4, 1008.5, 1013.7, 1018.8, 1024.0, 1029.2, 1034.4, 1039.6, 1044.8, 1050.0
3050 DATA 1055.2, 1060.4, 1065.6, 1070.8, 1076.1, 1081.3, 1086.6, 1091.9, 1097.2, 1102.4
3100 DATA 1107.7, 1113.0, 1118.3, 1123.7, 1129.0, 1134.3, 1139.7, 1145.0, 1150.4, 1155.8
3150 DATA 1161.2, 1166.6, 1172.0, 1177.4, 1182.9, 1188.3, 1193.8, 1199.2, 1204.7, 1210.2
3200 DATA 1215.7, 1221.2, 1226.8, 1232.3, 1237.9, 1243.5, 1249.1, 1254.7, 1260.3, 1265.9
3250 DATA 1271.6, 1277.3, 1282.9, 1288.6, 1294.3, 1300.1, 1305.8, 1311.5, 1317.3, 1323.1
3300 DATA 1328.9, 1334.7, 1340.5, 1346.4, 1352.2, 1358.1, 1363.9, 1369.8, 1375.7
3305 DIM TK(NK-1)
3310 FOR I = 0 TO NK-1:READ TK(I):NEXT I
3500 DIM DES$(7) : DIM DES1$(4)
3550 DATA Amb. temp.(c)      -,Manifold temp.(c) -,Exhaust temp.(c) -,Water temp.(c)-,Oil
temp.(c)      -,Inlet air temp.(c)-,Fuel temp.(c)-,Torque mom.(Nm) -
3600 FOR I=0 TO 7 : READ DES$(I) : NEXT I
3650 DATA Air-flow (g/s)      -, Fuel-flow(ml/s)-,Engine-speed(rpm)-
3700 FOR I=2 TO 4 : READ DES1$(I) : NEXT I
4000 GOSUB 45400 'Pick up the imformation charter
4100 PRINT "What kind of fuel are you using?"
4105 PRINT"<1>-Ethanol, maximun fumigation level"
4110 PRINT"<2>-Ethanol, minimun fumigation level"
4115 PRINT"<3>-Methanol, maximun fumigation level"
4120 PRINT"<4>-Methanol, minimun fumigation level"
4125 INPUT "Enter seletion number(1-4):";PAR%
4130 IF PAR%=1 THEN 4146
4135 IF PAR%=2 THEN GOSUB 46000
4140 IF PAR%=3 THEN GOSUB 47000
4145 IF PAR%=4 THEN GOSUB 48000
4146 ISTEP=3 : JSTEP=5
4150 PORT%(1)=956
4200 PORT%(2)=956
4250 PORT%(3)=958
4300 '
4350 PHPAT%(1,0)=3
4400 PHPAT%(1,1)=6
4450 PHPAT%(1,2)=12
4500 PHPAT%(1,3)=9
4550 '
4600 PHPAT%(2,0)=3 * 16
4650 PHPAT%(2,1)=6 * 16
4700 PHPAT%(2,2)=12* 16
4750 PHPAT%(2,3)= 9* 16
4800 '
4850 PHPAT%(3,0)=8
4900 PHPAT%(3,1)=13
4950 PHPAT%(3,2)=7
5000 PHPAT%(3,3)=2
5050 PHPCY%=4
5075 MOT1=0
5100 '-----
5150 '---Step 2: Check/Change the engine/test Set-up -----
5200 '-----
5250 PRINT:PRINT:PRINT:COLOR 0,7:INPUT" - Do you wish to check/change the engine/test setup(y/n)
- ";A$
5300 COLOR 1,7
5350 GOSUB 34800
5400 STAPOS%(2)=0
5450 CURPOS%(2)=STAPOS%(2)
6050 PRINT:PRINT:PRINT:COLOR 0,7:INPUT" - Do you wish to start/continue DASH-8 conversions
-(y/n)";A$
6100 COLOR 1,7
6150 IF A$ = "y" OR A$ = "Y" THEN 6350
6200 IF A$ = "n" OR A$ = "N" THEN CLS:IF CURPOS%(1)=0 THEN END
6450 IF LAB%=1 GOTO 6750
6750 CLEARANCE=SWEPT/(CR-1)

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6800 EPS=.5*STROKE/LENGTH
6850 PI=4*ATN(1)
6900 '
6950 '-----
7000 '---Step 3: Dimension data array for A/D data and check sufficient memory ---
7050 '-----
7100 'Check there is enough memory to hold this array
7150 IF (FRE(0) - 2000 - NC*2) < 0 THEN PRINT "Warning! There is inadequate memory within BASIC
to hold this data":PRINT"Re-run program":END
7200 DIM A%(150),PRESS(150),AVPR(150),AVPRM(150),VOL(150),THETA(150),B%(30)
7250 DIM E%(7),E1%(4),CH(7),CH1(4),AV(7,10),STD(7,10),AV1(4,10),STD1(4,10)
7300 DIM CHM(7),CHM1(4),CHA(7),CHA1(4)
7350 '-----
7400 '---Step 4: Transient Measurements -----
7450 '-----
7500 CLS: PRINT" To reach the Initial-Position, ":COLOR
0,7:LOCATE 1,50:PRINT"Press key F1":COLOR 1,7
7600 PRINT:PRINT:PRINT:COLOR 0,7:LOCATE 7,50:PRINT
7650 PRINT:PRINT:PRINT" To start the Steady-State measurements, ":COLOR 0,7:LOCATE
10,50:PRINT"Press key F4":COLOR 1,7
7700 PRINT:PRINT:PRINT" To start the Pressure-Volume measurements, ":COLOR 0,7:LOCATE
13,50:PRINT"Press key F5":COLOR 1,7
7750 LOCATE 25,1:COLOR 0,7:PRINT"F1"
7800 LOCATE 25,4:COLOR 1,7:PRINT"-In.Po. "
8050 LOCATE 25,39:COLOR 0,7:PRINT"F4"
8100 LOCATE 25,42:COLOR 1,7:PRINT"-St.St. "
8150 LOCATE 25,49:COLOR 0,7:PRINT"F5"
8200 LOCATE 25,52:COLOR 1,7:PRINT"-Pr.Vo. "
8250 KEY(1) ON : KEY(3) ON : KEY(4) ON : KEY(5) ON
8300 IF (CURPOS%(2) <> 0) AND (LAB%=1) THEN NDT%=500
8350 '
8400 ON KEY(1) GOSUB 39400
8500 ON KEY(3) GOSUB 40050
8550 ON KEY(4) GOSUB 40400
8600 ON KEY(5) GOSUB 40700
8650 '
8750 '
8800 GOTO 8750
13050 '-----
13100 '---Step 5: Get compensation temperature -----
13150 '-----
13151 C=0
13200 '
13250 MD%=1 : D%(0)=7 : D%(1)=7
13300 CALL DASH8(MD%,D%(0),FLAG%)
13350 IF FLAG% <> 0 THEN PRINT "error in setting ch 7":END
13400 MD%=4
13450 CALL DASH8(MD%,CJC%,FLAG%)
13500 IF FLAG% <> 0 THEN PRINT "error in setting ch 7":END
13550 CJC=CJC%/10
13600 '-----
13650 '---Step 6: EXP-16 Measurements -----
13700 '-----
13702 S=0 : JT=0
13703 PRINT "S=";S
13750 FOR SUB%= 0 TO 7 'note use of integer index SUB%
13800 LT%(0) = 0 : LT%(1) = 0 : MD% = 1
13850 CALL DASH8 (MD%, LT%(0), FLAG%)
13900 IF FLAG% <> 0 THEN PRINT "ERROR IN SETTING CHANNEL" : END
13950 'Next select each EXP-16 channel in turn and convert it.
14000 'Digital outputs OP1-4 drive the EXP-16 sub-multiplexer address, so use
14050 'mode 14 to set up the sub-multiplexer channel.
14100 J=SUB%
14101 IF J=6 GOTO 16050
14150 DEG%=NAV

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14200 FOR I=1 TO DEG%
14250 MD% = 14
14300 CALL DASH8 (MD%, SUB%, FLAG%)      'address set
14350 IF FLAG% <> 0 THEN PRINT "ERROR IN EXP-16 CHANNEL NUMBER" : END
14400 'Now that channel is selected, perform A/D conversion using mode 4.
14450 'Transfer data to corresponding array element e%(SUB%)
14500 MD% = 4 'do 1 A/D conversion
14550 CALL DASH8 (MD%, E%(SUB%), FLAG%)
14600 IF FLAG% <> 0 THEN PRINT "ERROR IN PERFORMING A/D CONVERSION"
14650 '      V = Exp-16 ch. voltage in volts
14700 V=E%(J)/GAIN*5/2048
14750 IF J>6 GOTO 15550
14800 '----- Interpolate to find K thermocouple temperature -----
14850 'Entry variables:-
14900 '      CJC = cold junction compensator temperature in deg. C.
14950 'Exit variables:-
15000 '      ch(J)= temperature in degrees Centigrade
15050 'Execution time on std. IBM P.C. = 46 milliseconds
15100 'Perform CJC compensation
15150 V0 = 1000*V + 1! + (CJC-25)*.0405      'V0 in mV
15200 'Find look up element
15250 EK = INT((V0-SVK)/SIK)
15300 IF EK<0 THEN CH(J)=TK(0):GOTO 15500 'Out of bounds, round to lower limit
15350 IF EK>NK-2 THEN CH(J)=TK(NK-1):GOTO 15500 'Out of bounds,round to upper limit
15400 'Do interpolation
15450 CH(J) = TK(EK) + (TK(EK+1) - TK(EK))*(V0-EK*SIK-SVK)/SIK      'Centigrade
15500 GOTO 15600
15550 CH(J)=(-V*1000)*FTORQ-12.9
15600 IF I=1 THEN AV(J,I)=CH(J):STD(J,I)=0
15650 IF I>1 THEN AV(J,I)=1/I*((I-1)*AV(J,I-1)+CH(J))
15700 IF I>1 THEN STD(J,I)=SQR((I-2)/(I-1)*STD(J,I-1)^2+I*(AV(J,I)-AV(J,I-1))^2)
15750 IF AV(J,I)=0 THEN GOTO 15850
15800 IF I=NAV THEN STD(J,I)=STD(J,I)*100/AV(J,I)
15850 NEXT I
15950 CHA(J)=(AV(J,NAV)-CHM(J))/AV(J,NAV)*100
16000 CHM(J)=AV(J,DEG%)
16050 NEXT SUB%
16100 '-----
16150 '---Step 7: STA-8 ( Channels 2 to 7 ) Measurements-----
16200 '-----
16250 MD% = 1      'Set scan limits, mode 1
16300 LL%=3 : D%(0)=LL%
16350 UL%=4 : D%(1)=UL%      'sample on two channels only:ch. 3 to 4
16400 NN=NAV*(UL%-LL%+1)
16450 CALL DASH8 (MD%, D%(0), FLAG%)
16500 IF FLAG% <> 0 THEN PRINT "Error in setting channel scan limits":STOP
16550 MD% = 5      'Mode 5, do conversions direct to array
16600 D%(0) = VARPTR(B%(0))      'Starting location of array
16650 D%(1) = NN      'Number of conversions
16700 CALL DASH8 (MD%, D%(0), FLAG%)
16750 IF FLAG% <> 0 THEN PRINT "Error in setting mode 5":STOP
16800 '
16850 FOR J=3 TO 4
16900 FOR I=1 TO DEG%
17000 I3=2*I-2
17050 I4=2*I-1
17075 CH1(3)=B%(I3)*5/2048*CFUEL-.15
17400 CH1(4)=B%(I4)*5/2048*FRPS
17450 IF I=1 THEN AV1(J,I)=CH1(J):STD1(J,I)=0
17500 IF I>1 THEN AV1(J,I)=1/I*((I-1)*AV1(J,I-1)+CH1(J))
17550 IF I>1 THEN STD1(J,I)=SQR((I-2)/(I-1)*STD1(J,I-1)^2+I*(AV1(J,I)-AV1(J,I-1))^2)
17600 IF AV1(J,I)=0 THEN GOTO 17700
17650 IF I=NAV THEN STD1(J,I)=STD1(J,I)*100/AV1(J,I)
17700 NEXT I
17750 IF DEG%=1 THEN 17850

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17800 CHA1(J)=(AV1(J,NAV)-CHM1(J))/AV1(J,NAV)*100
17850 CHM1(J)=AV1(J,DEG%)
17900 NEXT J
17901 AV1=AV1(4,DEG%) : AV2=AV1(3,DEG%)
17903 GOTO 18050
17950 IF LCHECK=1 GOTO 19500
18000 '-----
18050 '--- Step 8: Display results on screen -----
18100 '-----
18150 FOR I=1 TO 24:LOCATE I,1:PRINT " ";SPC(79):NEXT I
18200 LOCATE 1,1:PRINT"                                EXP-16 Channels: Average Results"
18250 PRINT"                                *****"
18300 PRINT" ch.no.      ch. description      value sta-dev      change in value      "
18350 PRINT" =====      =====      =====      =====      "
18400 FOR J=0 TO 7
18401 IF J=6 GOTO 18700
18450 PRINT USING" ##      ";J;
18500 PRINT USING" &";DES$(J);
18550 PRINT USING" ###.##";AV(J,DEG%);
18600 PRINT USING" ###.##";STD(J,NAV);
18650 PRINT USING" ###.##";CHA(J)
18700 NEXT J
18750 PRINT:PRINT:PRINT"                                STA-8 Channels: Average Results"
18800 PRINT"                                *****"
18850 PRINT" ch.no.      ch. description      value sta-dev      change in value      "
18900 PRINT" =====      =====      =====      =====      "
18950 FOR J=3 TO 4
19000 PRINT USING" ##      ";J;
19050 PRINT USING" &";DES1$(J);
19100 PRINT USING" ####.##";AV1(J,DEG%);
19150 PRINT USING" ###.##";STD1(J,NAV);
19200 PRINT USING" ###.##";CHA1(J)
19205 NEXT J
19208 '-----
19210 '---Step 9: Control of stepping motor and Heater-----
19212 '-----
19220 IF S<>0 THEN 19250
19230 GOSUB 43000 'Pick up the data for step-motor and heater
19240 GOSUB 45000 'Turn on the step-motor
19250 T=TI-AV(1,DEG%)
19260 PRINT "DELTA T=";T
19270 IF T>0 THEN PRINT "Raising the power of heater" ELSE PRINT "Decreasing the power of the
heater"
19300 LDIM=1
19351 IF JT<3 THEN 19355
19352 IF ABS(T)<3 THEN PRINT "Required temperature has been reached !!": GOSUB 40400
19355 '---- Set timer rate -----
19360 MD% = 10 'Mode 10 for setting counter configuration
19365 D%(0) = 2 'Operate on counter #2
19370 D%(1)=1 'Configuration #1 = Programmable one-shot
19375 CALL DASH8 (MD%, D%(0), FLAG%)
19380 IF FLAG% <> 0 THEN PRINT "Error in setting counter 2 configuration":STOP
19405 IF N<2 OR N >65535! THEN PRINT"Warning! A sample rate of ";F;" sample/sec is outside the
range of Counter 2":GOTO 19400
19410 MD% = 11 'Mode 11 to load counter
19415 D%(0) = 2 'Operate on counter #2
19420 IF N<32767 THEN D%(1) = N ELSE D%(1)=N-65536! 'Correct for interger
19425 CALL DASH8 (MD%, D%(0), FLAG)
19430 IF FLAG%<>0 THEN PRINT "Error in loading counter 2":STOP
19435 IF ABS(T)<3 THEN JT=JT+1 : GOTO 19450
19436 IF ABS(T)>10 THEN N=N-T*90 ELSE N=N-T*30
19450 S=1
19455 PRINT N: GOTO 13703
19498 '-----
19500 '--- Step 10: Pressure-Volume Measurements -----

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19550 '-----
19600 MD% = 1          'Set scan limits, mode 1
19650 LL%=1 : D%(0)=LL%
19700 UL%=LL% : D%(1)=UL% 'sample on one channel only:channel--1
19750 CALL DASH8 (MD%, D%(0), FLAG%)
19800 IF FLAG% <> 0 THEN PRINT "Error in setting channel scan limits":STOP
19850 'Note: Counter 2 output (pin 6) should be jumpered to interrupt input
19900 '(pin 24).
19950 FOR J=1 TO PVNO
20000 MD% = 5          'Mode 5, do conversions direct to array
20050 D%(0) = VARPTR(A%(0)) 'Starting location of array
20100 D%(1) = NC        'Number of conversions
20150 CALL DASH8 (MD%, D%(0), FLAG%)
20200 IF FLAG% <> 0 THEN PRINT "Error in setting mode 5":STOP
20250 '
20300 MINP=0
20350 FOR I=0 TO NC
20400 IF A%(I) < MINP THEN MINP=A%(I)
20450 THETA(I)=(I*360/(NC/2)+FTCA)/57.3
20500 SINT2=SIN(THETA(I))^2
20550 COST=1-COS(THETA(I))
20600 EPSIN=SQR(1-EPS^2*SINT2)
20650 EPSIN1=1/EPS*(1-EPSIN)
20700 VOL(I)=CLEARANCE*(1+.5*(CR-1)*COST+EPSIN1)
20750 PRESS(I)=(A%(I)-MINP)*5/2048*FPRESS
20800 IF J=1 THEN AVPR(I)=PRESS(I)
20850 IF J>1 THEN AVPR(I)=1/J*((J-1)*AVPRM(I)+PRESS(I))
20900 AVPRM(I)=AVPR(I)
20902 PRINT THETA(I)*57.3,    PRESS(I)
20950 NEXT I
21000 NEXT J
21050 '-----
21100 '--- Step 11: Calculate & display engine performance parameters -----
21150 '-----
21200 IMEP=0
21250 FOR I=0 TO NC-1
21300 IMEP=IMEP+.5*(AVPR(I)+AVPR(I+1))*(VOL(I+1)-VOL(I))
21350 NEXT I
21400 IMEP=IMEP/SWEPT
21800 BMEP=4*PI*AV(7,DEG%)/SWEPT/100
21850 FMEP=IMEP-BMEP
21900 BP=2*PI*AV(7,DEG%)*AV1/1000/60
22100 CLS:PRINT"                               Engine Performance Parameters    "
22150     PRINT"                               *****                               "
22200 PRINT" engine speed(rpm) = ";AV1
22300 PRINT" torque(Nm)         = ";AV(7,DEG%)
22350 PRINT" brake power(Kw)    = ";BP
22450 PRINT" bmep(bars)         = ";BMEP
22700 '-----
22750 '--- Step 12: Generate a Lotus 1-2-3 .PRN import file -----
22800 '-----
22850 LOCATE 25,1:INPUT"RESULTS.PRN FILE NAME [DRIVE]:NAME (automatic .PRN ext.) : ";F$
22900 F$ = F$ + ".PRN"
22950 OPEN F$ FOR OUTPUT AS #1
22951 GOTO 23250
23000 PRINT #1," theta(deg)   press(bars)   vol(liters)"
23050 PRINT #1," =====   =====   ====="
23100 FOR I = 0 TO NC
23150 PRINT #1, STR$(THETA(I)*57.3),STR$(PRESS(I)),STR$(VOL(I))
23200 NEXT I
23250 PRINT #1,"                               Exp-16 Channels: Average Results"
23300 PRINT #1,"                               *****"
23350 PRINT #1," ch.no.          ch. description          value          sta-dev(in %)"
23400 PRINT #1," =====          =====          =====          ====="
23450 FOR J=0 TO 7

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23451 IF J=6 GOTO 23550
23500 PRINT #1,J,DES$(J),AV(J,NAV),STD(J,NAV)
23550 NEXT J
23600 PRINT #1,"                               Sta-8 Channels: Average Results"
23650 PRINT #1,"                               *****"
23700 PRINT #1," ch.no.           ch. description           value           sta-dev(in %)"
23750 PRINT #1," =====           =====           =====           ====="
23850 PRINT #1,4,DES1$(4),AV1,STD1(4,NAV)
23950 PRINT #1,"                               Engine Performance Parameters           "
24000 PRINT #1,"                               *****           "
24050 PRINT #1," engine speed(rpm) = ";AV1
24150 PRINT #1," torque(Nm)           = ";AV(7,NAV)
24200 PRINT #1," brake power(Kw)      = ";BP
24300 PRINT #1," bmep(bars)           = ";BMEP
24550 CLOSE #1
24560 GOTO 7350
24600 '-----
24650 '--- Step 13: Generate DASH-8 graphics package data file -----
24700 '-----
24750 LOCATE 25,1:PRINT SPC(65)
24800 LOCATE 25,1:INPUT" PLOT.EXT FILE NAME [DRIVE]:NAME (automatic .EXT ext.) : ";FILX$
24850 FILX$=FILX$ + ".EXT"
24900 OPEN FILX$ AS #1 LEN = 30
24950 FIELD #1, 15 AS X$, 15 AS Y$
25000 'Enter number of data points and plot mode in record 1
25050 LSET X$ = MKS$(NC)
25100 PUT #1,1
25150 LSET Y$="PRESS  BARS"
25200 'Y axis label
25250 LSET X$="VOLUME(LITERS)"
25300 'X axis label
25350 PUT #1,2
25400 'Enter data in remaining records
25450 FOR I=3 TO NC+3
25500 LSET X$=MKS$(VOL(I-2)):LSET Y$=MKS$(PRESS(I-2))
25550 PUT #1, I
25600 NEXT I
25650 'Write file
25700 CLOSE #1
25750 '-----
25800 '--- Step 14: Run plotting subroutine -----
25850 '-----
25950 OPEN "o",#2,"motpos.dat"
26050 PRINT #2,CURPOS%(1)
26150 RUN"plot
26200 END
26250 '
33950 ' Subroutine for controlling the step-motors
34000 ' *****
34050 '
34100 IF DOM >= REVMAX THEN DOMM=DOM : RETURN
34150 IF (DOM > 1) AND (DOM <= REVMIN) THEN DOMM=DOM : RETURN
34200 GRAD%=0
34250 IF DOM <> DOMM THEN GRAD%=CINT(KOPT*(NOM-NOMM)/(DOM-DOMM))
34300 IF GRAD%=0 THEN RETURN
34350 IF GRAD% > .2*CURPOS%(MOTNU%) THEN GRAD%=.2*CURPOS%(MOTNU%)
34400 ENDPOS%(MOTNU%)=CURPOS%(MOTNU%)+GRAD% : GOSUB 40850
34450 FLG%=FLG%+1
34500 'print flg%,nom,nomm,dm,domm,grad%,motnu%,curpos%(motnu%)
34550 NOMM=NOM : DOMM=DOM : RETURN
34600 '
34650 ' Subroutine for checking/changing the engine/test set-up
34700 ' *****
34750 '
34800 DIM A(27),B$(27)

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34850 OPEN "i",#1,"setupt.dat"
34900 FOR I=1 TO 27
34950 B$(I)=INPUT$(26,#1)
35000 INPUT #1,A(I)
35050 NEXT I
35100 CLOSE #1
35150 IF A$ = "y" OR A$ = "Y" THEN GOTO 35250
35200 IF A$ = "n" OR A$ = "N" THEN GOTO 36350
35250 CLS: HED$="Engine/Test specifications set-up"
35300 TB%=(80-LEN(HED$))/2:PRINT TAB(TB%) HED$
35350 PRINT TAB(TB%)"*****"
35400 PRINT
35450 FOR I=1 TO 27
35500 IF I<=14 THEN PRINT I;B$(I);A(I)
35550 IF I>14 THEN LOCATE I-11,40:PRINT I;B$(I);A(I)
35600 NEXT I
35650 LOCATE 20,1:PRINT"Choose I: No. of parameter to be changed."
35700 LOCATE 21,1:PRINT"(To leave the set-up,press the <Enter> key)";
35750 PS=POS(0):PRINT "      ":LOCATE 21,PS:BEEP
35800 LINE INPUT N$:IF N$="" THEN 36350
35850 I=VAL(N$)
35900 IF I>14 THEN ROW=I-11:COL=70
35950 IF I<=14 THEN ROW=I+3:COL=30
36000 IF I>27 THEN 35700
36050 LOCATE ROW,COL:COLOR 24,7:PRINT A(I)
36100 LOCATE ROW,COL+1:COLOR 1,7:LINE INPUT X$
36150 IF X$<>"" THEN A(I)=VAL(X$)
36200 LOCATE ROW,COL:PRINT A(I)
36250 LOCATE ROW,COL+LEN(STR$(A(I))):PRINT SPC(10-LEN(STR$(A(I))))
36300 GOTO 35700
36350 OPEN "o",#2,"setupt.dat"
36400 'B$( 1)="STROKE LENGTH(M)      ="
36450 'B$( 2)="CONNECTING ROD LENGTH(M) ="
36500 'B$( 3)="BORE LENGTH(M)        ="
36550 'B$( 4)="MISAL. VOLUME(LITER)   ="
36600 'B$( 5)="SWEPT VOLUME(LITER)    ="
36650 'B$( 6)="COMPRESSION RATIO      ="
36700 'B$( 7)="INJEC. START - ATDC(DEG) ="
36750 'B$( 8)="INLET OPENING TIME(DEG) ="
36800 'B$( 9)="VOL. FUEL RATE(CC/S)    ="
36850 'B$(10)="LCV(MJ/KG)              ="
36900 'B$(11)="A/F STOIC.              ="
36950 'B$(12)="FUEL DENSITY(G/CC)      ="
37000 'B$(13)="EXP-16'S GAIN          ="
37050 'B$(14)="NO. OF AVE. SAMPLINGS  ="
37100 'B$(15)="NO. OF P-V DIAGRAMS    ="
37150 'B$(16)="NO. OF P-V SAMPLS./CYCL. ="
37200 'B$(17)="TORQUE FAC.(NM/V)      ="
37250 'B$(18)="REV. COUNTER FAC.(RPS/V) ="
37300 'B$(19)="CYL. PRESS. FAC.(BAR/V) ="
37350 'B$(20)="INJEC. TIME FAC.(MSEC/V) ="
37400 'B$(21)="INJEC. TIME OFF.(MSEC)  ="
37450 'B$(22)="CONTROLLED LAMDA       ="
37500 'B$(23)="CONTROLLED SPEED       ="
37550 'B$(24)="MAXIMUM SPEED(RPS)     ="
37600 'B$(25)="MINIMUM SPEED(RPS)     ="
37650 'B$(26)="1st MOTOR In.Po.(mt.sts) ="
37700 'B$(27)="2nd MOTOR In.Po.(mt.sts) ="
37750 STROKE=A(1)
37800 LENGTH=A(2)
37850 BORE=A(3)
37900 MISAVOL=A(4)
37950 SWEPT=A(5)
38000 CR=A(6)
38050 FTCA=A(7)

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38100 INLANG=A(8)
38150 VDFUEL=A(9)
38200 LCV=A(10)
38250 AFSTO=A(11)
38300 DENSFU=A(12)
38350 GAIN=A(13)
38400 NAV=A(14)
38450 PVNO=A(15)
38500 NC=A(16)
38550 FTORQ=A(17)
38600 FRPS=A(18)
38650 FPRESS=A(19)
38700 CFUEL=A(20)
38750 OFFUEL=A(21)
38800 LAMDA=A(22)
38850 RPS=A(23)
38900 REVMAX=A(24)
38950 REVMIN=A(25)
39000 INIPOS%(1)=A(26)
39050 INIPOS%(2)=A(27)
39100 FOR I=1 TO 27
39150 PRINT #2,B$(I);A(I)
39200 NEXT I
39250 CLOSE #2
39300 RETURN 5400
39350 '
39400 ' KEY(1) Subroutine
39450 ' *****
39500 MOTNU%=2
39600 ENDPOS%(MOTNU%)=INIPOS%(MOTNU%)
39650 GOSUB 40850
39700 LOCATE 22,1:PRINT " Initial-Positions' been reached. Press":COLOR 24,7:LOCATE
22,42:PRINT"OTHER CHOICE":COLOR 1,7
39710 MOT1=0 : CURPOS%(2)=0
39750 RETURN 8800
40300 '
40350 '
40400 ' KEY(4) Subroutine
40450 ' *****
40500 TRANS%=1
40550 RETURN 13100
40600 '
40650 '
40700 ' KEY(5) Subroutine
40750 ' *****
40800 RETURN 19500
40850 '
40900 ' This subroutine moves three step motors from position 0
40950 ' *****
41000 '
41050 IF CURPOS%(MOTNU%) < ENDPOS%(MOTNU%) THEN STP% = 1 ELSE STP% = -1
41100 '
41150 IF CURPOS%(MOTNU%) = ENDPOS%(MOTNU%) THEN 41550
41200 CURPOS%(MOTNU%) = CURPOS%(MOTNU%) + STP%
41250 IF CURPOS%(MOTNU%) < 0 THEN PRINT "Negative position . Press <ESC> to continue":END
41300 PHNU%=PHPAT%(MOTNU%,CURPOS%(MOTNU%) MOD PHPCY%)
41350 OUT PORT%(MOTNU%),PHNU%
41400 FOR I=1 TO 200
41450 NEXT I
41500 GOTO 41150
41550 RETURN
41600 '
41650 '
41700 ' This subroutine removes the step motors to starting position
41750 ' *****

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41800 '
41850 CLS:LOCATE 5,35:COLOR 24,7:PRINT "WARNING"
41900 LOCATE 10,5:COLOR 1,7:PRINT "Before you finish the test"
41950 LOCATE 10,6:PRINT "be sure to press on key MOTORING on the control board"
42000 LOCATE 23,1:COLOR 1,7:PRINT "When you're ready --- press any key to continue"
42050 IF INKEY$="" GOTO 42050
42150 MOTNU%=2 : NDT%=500
42200 ENDPOS%(MOTNU%)=STAPOS%(MOTNU%)
42300 GOSUB 40850
42400 LOCATE 25,1:PRINT "Starting-Positions' been reached "
42450 FOR I%=1 TO 100
42500 NEXT I%
42550 END
43000 'Subroutine for picking up data of step-motor & heater
43025 *****
43050 I=1 : J=2
43100 DR=ABS(AV1-R(I))
43150 IF DR<90 THEN GOTO 43250
43200 I=I+1
43210 IF I=<ISTEP THEN 43100
43220 PRINT "Check the engine speed (rpm) "
43230 GOTO 7750
43250 DT=AV(7,DEG%)-T(I,J-1) : DT1=AV(7,DEG%)-T(I,J)
43300 IF DT>0 AND DT1<0 THEN GOTO 43400
43350 J=J+1
43360 IF J=<JSTEP THEN 43250
43370 PRINT "Check the strain bridge"
43380 GOTO 7750
43400 MOT=M(I,J-1)+(M(I,J)-M(I,J-1))/(T(I,J)-T(I,J-1))*DT
43410 TI=H(I,J-1)+(H(I,J)-H(I,J-1))/(T(I,J)-T(I,J-1))*DT
43415 MOT=.65*MOT
43420 MOT=INT(MOT) : N=N(I,J-1) : SETTING=MOT/5
43455 PRINT "M step-motor=";MOT;"T heater=";TI; "Step-motor setting="SETTING
43470 C=9999
43475 DMOT=MOT-MOT1 : PRINT "dMOT=";DMOT;
43500 RETURN
45000 'Subroutine to turn the step-motor for fumigation control
45025 *****
45050 MOTNU%=2 : COUNT%=0
45100 COUNT%=COUNT%+1
45150 IF COUNT%=17 THEN COUNT%=0
45200 ENDPOS%(MOTNU%)=CURPOS%(MOTNU%)+DMOT
45250 GOSUB 40850
45275 MOT1=MOT : PRINT "CURPOS%(2)=";CURPOS%(2), "MOT1=";MOT1
45300 RETURN
45400 'Subroutine of information data for computer control
45450 'data for ethanol, maximum fumigation
45475 *****
45500 DIM R(3),T(3,5),H(3,5),M(3,5),N(3,5)
45505 FOR I=1 TO 3 : FOR J=1 TO 5
45510 READ R(I),T(I,J),H(I,J),M(I,J),N(I,J)
45515 DATA 1000,2.5,65,57,30000,1000,8,58,64,30000,1000,15.7,48,70,37000
45520 DATA 1000,17.9,35,90,40000,1000,29.1,38,87,42000,1500,3.8,25,83,31000
45525 DATA 1500,10.5,25,90,31000,1500,19.7,21,95,32000,1500,28.1,25,89,34000
45530 DATA 1500,29.4,26,90,32500,2000,5,40,75,34000,2000,7.5,40,76,32000
45535 DATA 2000,12.8,40,78,29000,2000,18,39,68,35000,2000,23.7,38,57,40000
45565 NEXT J
45570 NEXT I
45572 'data for ethanol, minimum fumigation
45573 *****
45575 DIM R1(3),T1(3,5),H1(3,5),M1(3,5),N1(3,5)
45580 FOR I=1 TO 3 : FOR J=1 TO 5
45585 READ R1(I),T1(I,J),H1(I,J),M1(I,J),N1(I,J)
45590 DATA 1000,9.0,41,57,38000,1000,13,40,58,40000,1000,17.0,40,59,42000
45595 DATA 1000,21,40,52,42000,1000,25.6,40,45,42500,1500,3.86,41,50,41000

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45600 DATA 1500,6.9,41,32,42500,1500,11,41,21,40000,1500,16.2,41,10,37500
45605 DATA 1500,25.4,40,22,43000,2000,5.1,38,55,41000,2000,8.5,38,53,41500
45610 DATA 2000,12.5,38,58,42000,2000,17,42,40,40500,2000,22.1,43,29,39000
45620 NEXT J : NEXT I
45630 'data for methanol, maximun fumigation
45640 *****
45650 DIM R2(3),T2(3,5),H2(3,5),M2(3,5),N2(3,5)
45660 FOR I=1 TO 3 : FOR J=1 TO 5
45670 READ R2(I),T2(I,J),H2(I,J),M2(I,J),N2(I,J)
45680 DATA 1000,2.3,39,97,34000,1000,7,42,91,34000,1000,11.3,45,83,34000
45690 DATA 1000,19.5,43,83,40000,1000,29.0,48,62,42500,1500,3.8,36,95,38000
45700 DATA 1500,8.3,37,86,40000,1500,13,39,85,38500,1500,19.3,40,85,37500
45710 DATA 1500,28.3,41,73,44000,2000,0.2,41,5,47500,2000,4,37,45,47500
45720 DATA 2000,8.2,33,87,48000,2000,17.3,41,80,31000,2000,27.3,37,87,49000
45730 NEXT J : NEXT I
45740 'data for methanol, minimun fumigation
45750 *****
45760 DIM R3(3),T3(3,5),H3(3,5),M3(3,5),N3(3,5)
45770 FOR I=1 TO 3 : FOR J=1 TO 5
45780 READ R3(I),T3(I,J),H3(I,J),M3(I,J),N3(I,J)
45790 DATA 1000,2.3,50,56,34500,1000,7.5,48,54,34500,1000,10.9,46,53,35000
45800 DATA 1000,19.5,53,22,35000,1000,26.5,55,20,35000,1500,3.6,56,0,37000
45810 DATA 1500,8.2,56,0,37000,1500,15,55,0,37000,1500,20.2,53,0,37000
45820 DATA 1500,26.2,54,0,37000,2000,4.8,30,0,48000,2000,10,45,0,46000
45830 DATA 2000,17.3,55,0,45000,2000,22,46,30,47000,2000,27.3,37,87,49000
45840 NEXT J : NEXT I
45850 RETURN
46000 'Subroutine for ethanol array, minimun fumigation
46050 *****
46100 FOR I=1 TO 3 : FOR J=1 TO 5
46150 R(I)=R1(I) : T(I,J)=T1(I,J):H(I,J)=H1(I,J):M(I,J)=M1(I,J):N(I,J)=N1(I,J)
46200 NEXT J: NEXT I
46210 RETURN
47000 'Subroutine for Methanol array, maximun fumigation
47050 *****
47100 FOR I=1 TO 3 : FOR J=1 TO 5
47150 R(I)=R2(I) : T(I,J)=T2(I,J):H(I,J)=H2(I,J):M(I,J)=M2(I,J):N(I,J)=N2(I,J)
47200 NEXT J: NEXT I
47210 RETURN
48000 'Subroutine for Methanol array, minimun fumigation
48050 *****
48100 FOR I=1 TO 3 : FOR J=1 TO 5
48150 R(I)=R3(I) : T(I,J)=T3(I,J):H(I,J)=H3(I,J):M(I,J)=M3(I,J):N(I,J)=N3(I,J)
48200 NEXT J: NEXT I
48210 RETURN

```

APPENDIX C

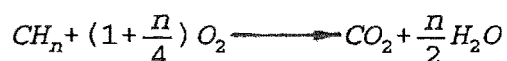
POLLUTANT EMISSIONS FROM CI ENGINES

Since the alarm that was caused by the photochemical smog cloud, that hung over the Californian basin in the late 1960's, global concern about health and environmental impact has been rising. This is especially true in urban areas where pollution from vehicle engines has caused the introduction of tough legislations to limit pollutant levels in a campaign to provide cleaner air.

Today there is clean air legislation in North America, Europe and Japan. New Zealand is one of the few industrialised countries which has not legislated to control the level of exhaust emissions. Due to the scale of transportation and the geographical advantage of New Zealand, pollution has not yet reached the level of becoming a major environmental problem. New Zealand must however remain aware of potential problems and monitor the matter accordingly.

C.1. FORMATION OF POLLUTANT EMISSIONS

Diesel fuel consists of organic molecules which are mostly hydrocarbons. When such compounds completely burn in oxygen the following generalized reaction occurs:



(Where n is the hydrogen carbon ratio of the fuel).

In reality, combustion is not complete and partially oxidized products are produced such as carbon monoxide,

lower order hydrocarbons and oxygen containing species such as aldehydes. Detail can be found in Table C1 (50).

Exhaust gas constituents		At idle	At Max. power
Oxides of nitrogen	Vol%	.005-.0025	.06-.15
Hydrocarbons	Vol%	.05-.06	.02-.06
Carbon monoxide	Vol%	.01-.045	.035-.2
Carbon dioxide	Vol%	3.5	12.0
Steam	Vol%	3.0	11.0
Oxygen	Vol%	16.0	10.0
Nitrogen etc.	Vol%	remainder	remainder
Soot	mg/m ³	20	200
Exhaust temperature	°C	100-200	550-750

Table C1. Exhaust gas composition and temperature.

These exhaust emission parameters were not measured when diesel engine was operated on alcohol fuels using the fumigation method. However, the following information is provided to assist with an understanding of problem.

C.2. PARTICULATE MATTER

Particulate emissions (soot) in diesel engine is high compared to SI engines. They consist of a mixture of carbon and heavy hydrocarbons together with traces of sulphuric acid arising from sulphur in the fuel. Diesel

engine output is limited by the visible smoke emissions. In general, maximum fuel quantity must be limited according to intake air quantity so that the engine produces little or no soot. This requires an excess-air factor of 10% - 20% ($\lambda=1.1 - 1.2$).

The carbon and hydrocarbons come from incomplete combustion of diesel fuel, resulting from the heterogeneous combustion occurring in the diesel engine. This is an order of magnitude higher than those of gasoline engines. Optimization of combustion chamber and careful choice of start of injection, injection sequence characteristics and fuel atomization have effects on the toxic emissions.

When a diesel engine is run on alcohol fuels, the smoke problem can be reduced significantly. Murayama et al (31) found that smoke was extinguished by the fumigation of ethanol. Research by Duggal et al (32) also found the smoke level with ethanol fumigation was lower than that with diesel below 60% maximum engine load.

C.3. CARBON MONOXIDE

Carbon monoxide results from incomplete combustion and obviously increases as the loading increases. This can vary from a few ppm at low load to 3000 ppm at high load where the air/fuel ration drops to 20:1 from the highest levels of around 100:1. The variation of carbon monoxide emission with air/fuel ratio and hence with load is shown in Figure C1.(38).

Usually a diesel engine operates with low smoke limits and therefore operates with excess air, hence carbon monoxide and carbon dioxide are also low. Diesel engines

therefore usually have little difficulty in meeting legislated levels.

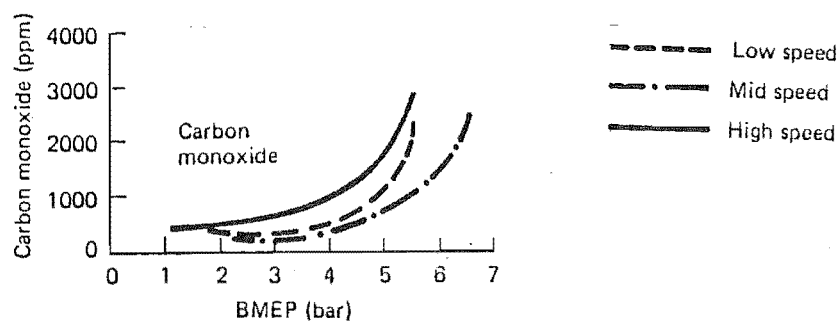


Figure C1 Relation between emission of CO and load for an IDI diesel engine.

C4. UNBURNT HYDROCARBON

Unburned or partially oxidised fuel may appear in the exhaust gases as hydrocarbons or oxygenated species such as aldehydes and ketone.

Excess unburnt hydrocarbon emissions mainly arise from shortcomings in the fuel injection system. They depend on load and speed but not on air/fuel ratio. Fuel also has an effect on hydrocarbon emissions. There is a trend to higher hydrocarbon emissions with lower end boiling point fuels.

Some higher hydrocarbon levels were noticed with fumigated alcohol. Foster et al (42) found that HC levels with fumigated ethanol were up to 12% higher than that with diesel operation in a DI engine, whilst the IDI engine had larger percentage changes in the HC emission than DI engine. Another fumigation test by Murayama et al (31) found that hydrocarbons at light loads may reach 1000 ppm, unburnt ethanol 2000 ppm and total aldehydes 400 ppm. Hence control is necessary to stop ethanol fumigation at the light loads.

C.5. NITROGEN OXIDES

Nitrogen oxide emissions are generated in the combustion chamber by two mechanisms as follow:



In both case, the nitrogen oxides are generated by the reaction of nitrogen and oxygen from the air, producing the toxic diesel emissions. With moisture and further oxygen (both are present in the lungs), they tend to form nitric acid, which is harmful to human body and is involved in the formation of photochemical smog.

The above mechanism is highly dependent on temperature, linearly dependent on oxygen atom concentration and independent of the fuel type. The final emission also tends to be proportional to post-temperature residence times.

By retarding the injection timing, useful reductions in nitric oxides emissions can be achieved for all diesel engines.

Foster et al (42) concluded that changes caused by the increased percentage of fumigated alcohol would result in lower temperatures and pressures at the start of injection which would tend to increase the ignition delay and cause slightly lower flame temperatures, with a subsequent decrease in NO_x emissions. The indirect injection engines tend to produce lower NO_x emissions because of the more retarded timings which are normally used.